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DEVELOPMENT OF AIRCRAFT SHOCK ABSORBERS USING FRICTION AS THE ENERGY DISSIPATOR

W. W. FRICKER

THE CLEVELAND PNEUMATIC TOOL CO.

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FOREWORD

The research work described in this report was performed by the Cleveland Pneumatic Tool Company, a subsidiary of Pneumo Dynamics Corporation, Cleveland, Ohio, for the Systems Engineering Group, Directorate of R & D Procurement. Aeromechanics Division, Flight Dynamics Branch, Wright-Patterson Air Force Base, Ohio, under Air Force Contract No. AF33(615)-2033.

This research is the initial part of a scale model study to investigate the use of Coulomb friction in the design of high temperature aircraft shock absorbers. This work was performed under Project No. 1369, Mechanical Subsystems for Aerospace Vehicles and the Task No. is 136903, Launching and Alighting Systems for Ground Contact.

The work under the above contract was monitored by the Air Force Flight Dynamics, Mechanical Branch, with Mr. Peter Smits as the project engineer and Mr. Aivars V. Petersons, Technical Manager. The research was conducted during the period of July 1964 to April 1965 by W. W. Fricker, contractor project engineer assisted by specialized engineering personnel from the Cleveland Pneumatic Tool Company.

The author wishes to acknowledge the participation in the program of the following organizations:

The Metal Bellows Corporation, R. E. Dearborn, for his continued support and assistance.

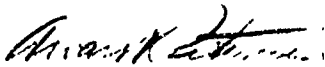
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This technical report has been reviewed and is approved.


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Acting Chief, Mechanical Branch
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ABSTRACT

The purpose of this study is to evolve and evaluate a number of preliminary aircraft landing gear designs or concepts utilizing shock absorbers based on friction energy absorption methods adaptable to high temperature use (1000° F and up).

Initial studies developed as many concepts as possible, without attempting to size components, but merely to portray an idea. Four (4) concepts were selected from the initial study and preliminary designs of the selected concepts were prepared based on landing and take-off characteristics of the USAF F-100 aircraft. Scale model drawings were then developed based on the full scale landing gear designs to experimentally evaluate the proposed designs.

Data indicates that friction methods have extremely high energy absorption capacities, that the coefficient of friction is relatively insensitive to temperature variations and that a centrifugally controlled friction device has sufficient advantages to warrant further research effort.

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SECTION I

INTRODUCTION

On 1 July 1964, a Contract No. AF33(615)-2033, was undertaken by the Cleveland Pneumatic Tool Co., to investigate, evolve and evaluate concepts for aircraft shock absorbers using friction material as the energy dissipator.

Existing aircraft landing gear designs are adversely affected by large, sudden temperature variations as well as by temperature extremes. This problem is of such magnitude that the existing designs are unsuited for projected future aircraft. For extreme cases, the present state of the art demands controlled temperature environment for the gear assembly. Among the difficulties associated with the temperature extremes are seals for either gases or liquids and hydraulic fluids with satisfactory properties over the temperature range.

Existing seals at temperatures below -65° F are not satisfactory. At the other end of the scale, metallic seals become necessary at temperatures much over 450° F and these present problems of leakage.

Fluids also have limited usable temperature ranges. Conventional hydraulic fluids become very viscous at -80° F and represent a serious fire hazard over 275° F. At higher temperatures, they break down and become extremely difficult to seal because of their low viscosity. The damping characteristics change with large temperature changes to the extent that lower shock absorber efficiencies must be assumed necessitating longer strokes or higher load factors, either of which add weight to the aircraft. Silicone fluids have an appreciably lower viscosity change over a specified temperature change. However, the problems still exist but at a lower magnitude.

Since it is highly unlikely that suitable oils and seals will be developed in the near future, an unconventional approach in the development of a high temperature shock absorber is warranted.

In view of the fact that friction methods have extremely high energy absorption capacities (in excess of 200,000 ~~FT-#~~) and that the coefficient of friction is relatively insensitive to temperature variation, the utilization of this technique in shock strut components offers promise.

Among the problems associated with the use of friction devices for landing gear applications are:

1. Significant spread between the static and the sliding coefficient of friction. This tends to produce initial loads which are higher than the load for the major part of the stroke. This results in low shock absorber efficiency. . . if not compensated.
2. If the device develops constant force during the sliding phase, as it would tend to do without control devices, the stroke would stop when the energy was dissipated. Thus, at low sinking speeds, the airplane would come to rest with the gears nearly extended and at high sinking speeds, with the gear compressed. This presents many problems in stability, angle of attack, and floor level with respect to the ground.
3. A spring for taxing is not built in as in the case of conventional aircraft shock absorbers.
4. A means of positioning the gear to the fully extended position for landing must be provided to work against or release the friction unit.
5. Provision must be made to insure that the coefficient of friction does not change due to temperature, rate of travel, presence of moisture, oil or grease, sand or other dirt, and wear of the friction materials. Otherwise, low shock absorber efficiency will result unless compensation is provided.
6. Loss in strength of components due to temperature rise if they serve as structural members as well as heat sinks.

The ability to use friction energy absorption methods at 1000°F and over does not present an insurmountable problem. The temperature rise in the braking system of high performance airplanes is such that peak temperatures up to 2500°F are encountered under rejected take-off conditions. The energy ratio is roughly 1 to 900. Thus, when starting at a temperature of 1000°F or somewhat higher, a relatively small heat sink is required, even though its temperature is high, to absorb the maximum vertical energy while staying within the present brake temperature limits.

Initially as many concepts as possible for utilizing friction energy absorption methods were derived, no matter how impractical, and were analyzed for their feasibility and value for further evaluation. From this list

approximately 25 concepts were rated on the basis of seven parameters using an emphasis curve to establish the relative weight for each of the criteria considered in the rating.

At the completion of this evaluation a meeting with the FDFM project engineer was held to select four concepts to be developed into scale model designs.

In addition to concept evaluation a limited literature survey was conducted to (a) discover whether any of the methods suggested have been previously utilized and (b) friction materials were surveyed for their applicability to the various designs.

The design phase of the project consisted of the preparation of preliminary drawings of the selected gear concepts based on landing and take-off characteristics of an actual high performance airplane. In this case the USAF F-100 aircraft was selected.

The modeling phase of the project consisted of selecting the proper dimensionless ratios and scale ratios to permit dynamic scaling of the full size preliminary designs. The final result of the program is a set of drawings of scale models designed to experimentally evaluate the proposed shock absorber designs.

REVERSE IS BLANK

SECTION II

CONCEPT DEVELOPMENT AND EVALUATION

A number of principles and concepts to solve the problems stated previously in the "Introduction" section of this report have been generated and are discussed below. Since one of the objectives of the project was to explore as many ideas as possible for utilizing energy absorption methods, the concepts and solutions to the problems which are disclosed herein are presented as the basic framework of parameters for concept formulation. They are presented as typical examples and are indicative of the original thinking which Cleveland Pneumatic applied to the entire project. Discussion of some of the parameters for concept formulation are presented in the following paragraphs.

A. STATIC VS. SLIDING FRICTION COEFFICIENT

The ratio between the static and sliding coefficient of friction of most suitable braking material combinations is appreciable, ranging from a value of 1.5:1 to 20:1, even though many satisfactory combinations have a ratio of 1.5:1. It is obvious that a simple device providing a constant force against the brake surfaces is not suitable because of the ensuing low shock absorber efficiencies which would result.

An obvious solution would be to find a combination of materials where the friction coefficient was constant regardless of the conditions. This was worthy of a significant search of the literature as well as contact with brake manufacturing people.

Assuming that no material combination gives a constant value of friction coefficient for static and the full velocity range of sliding conditions; other methods of achieving near constant loads were developed. The simple case would be represented by a brake attached to a cylinder which clamped around a piston that stroked in the cylinder. If the piston outside diameter is reduced within an inch of travel, the problem of breakaway friction is eliminated. While this may be an oversimplification of the problem, it does demonstrate that it can be overcome. Another example which is sketched and shown in Figure 32 uses a rack attached to the moving member which causes brake discs to rotate when relative motion exists between two members of the shock absorber system. To reduce breakaway forces a cam surface is provided on the rack member, upon which a roller follower rides. Motion of the follower applies the braking force. Thus, as in the previous case, there is little frictional force at breakaway and essentially constant force after a significant velocity is reached.

B. FINAL STROKE POSITION

The discussion above dealt with methods of developing essentially constant forces during the stroke. Ideally, nearly full stroke would be used with the force being essentially constant at a magnitude just large enough to dissipate all of the energy put into the shock absorber. The vertical load then would be approximately proportional to the square of the sinking speed. The advantages from a fatigue standpoint are obvious.

An additional advantage which would result from a unit that always uses practically full stroke is a uniform static position for taxi and takeoff. This puts the center of gravity close to the ground for stability in taxiing, establishes a definite airplane attitude for takeoff, and provides a specific distance from the airplane floor to the ground.

Figure 33 shows a method of approximately achieving constant force for the full stroke regardless of the sinking speed. A centrifugal type of governor is driven by the rack through step-up gearing. The geometry of the governor and the connecting linkage which applies the braking force is established to relate the braking force to the sinking speed and attempt to produce a flat load vs. stroke curve. Realizing that the ideal will not be accomplished, the brake is adjusted so that there is always a minimum braking force of slightly less than the minimum static reaction on the gear. Then, even though the kinetic energy is dissipated before full stroke is reached, the unit will stroke to the static position.

C. TAXIING SPRING

Taxiing conditions make a spring in the system mandatory for high performance airplanes of the B-58, F-104 and F-105 types. The nature of the "tire" and wheel assembly characteristics has bearing upon the requirements of the landing gear. If sufficient deflection is available in the wheel assembly to accommodate the taxiing conditions, no spring travel in the shock absorber is necessary. However, assuming that spring deflection is necessary in the shock absorber, it appears that the articulated arrangement shown in Figure 30, with the beam to which the wheel is attached acting as a cantilever spring, would serve very well when so designed. This principle could be employed for either main or nose gears.

D. GEAR EXTENSION

The discussion under "Final Stroke Position" above indicates the desirability of a spring loaded friction device which produces a force slightly less than the minimum static reaction on the gear after the kinetic energy is dissipated. It is necessary, however, for the gear to extend itself on take-off so as to have available stroke for the next landing impact. Friction would prevent the unsprung weight of the gear from extending the unit. Therefore,

a brake release mechanism is needed to eliminate the brake friction when the wheel is not touching the ground. This can be accomplished by a free wheeling clutch type of unit, for example, or by a brake release device.

E. CONSTANT COEFFICIENT OF FRICTION

The performance of automotive brakes in which the presence of moisture or grease usually makes the brake grab, is indicative of a serious problem for a friction type of shock absorber. Other environmental conditions such as extreme temperature and sand and dust are expected to have significant influence upon the performance. It is quite apparent that the ideal design condition exists when the load under a specified energy input is known within narrow limits. With the coefficient of friction varying with the environment the load is uncertain and the gear efficiency is lower than desired. As a result, means of sealing or protecting the friction assembly from the troublesome environment or eliminating the effect is necessary. Sealing from the entrance of foreign matter is possible.

F. HIGH TEMPERATURE STRENGTH LOSS

Structural materials generally have lower strength at elevated temperatures than at room temperature. Starting braking action at 1000°F or over eliminates many materials which might otherwise be used. This is particularly critical in a design which uses the structural member for the braking surface. Then, the high loads and the high temperature would occur practically simultaneously. It appears logical to utilize an independent brake which does not require great strength to carry the landing loads, as illustrated by Figures 11 and 12. Further studies should be to determine; whether beef-up to a structural member to compensate for strength loss due to heating is less efficient than a separate heat sink.

G. PROBLEM AREAS

1. Static vs. Sliding Friction Coefficient

The difference between the static and the sliding coefficient of friction has been discussed previously. A combination of usable friction materials which will have a constant friction coefficient is not likely to be found. A mechanical means of overcoming the high breakaway friction has an excellent chance of being successful.

2. Friction Coefficient Change with Environment

The effect of various environments upon the friction coefficient may present a serious problem when striving for highly efficient energy absorbers. Among the solutions to the problem are:

- 1.) Sealing the friction assembly from the environment.
- 2.) Selecting materials which do not exhibit friction coefficient changes.
- 3.) Designing for lower efficiency of the shock absorber.

Sealing of the assembly from the environment is practical. The new high performance wheel brakes perform very satisfactorily even when tested for the rejected take-off condition where the brakes reach 2500° F. Thus, sealing of the brake assembly from foreign matter may be all that is needed. Materials that have a constant friction coefficient regardless of the environment are not likely to be found. However, the program certainly demands a search for such a combination. Obviously, if all other possible solutions fail, the assembly can be designed for low efficiency. This approach would add weight and should be considered as a last resort.

H. RATING SYSTEM

One of the most difficult problems in the objective evaluation of competing systems is the establishment of relative weights for the various criteria of interest. It is not usually difficult to decide what the criteria should be, but any attempt to consider a large number of factors simultaneously and keep their relative importance in mind is almost certain to be confusing. One approach to the development of a more truly objective rating system is the use of an emphasis curve. This approach was used for the Friction Shock Absorber rating system. The method is fairly straight forward, and may be outlined in a very few steps.

1. List the criteria to be considered, with no regard to their relative importance. Assign an arbitrary number (in sequence) to each factor.
2. Establish a matrix which pairs each factor with each other factor, as follows (example shown for 7 factors):

1	1	1	1	1	1	1	Factor Number	Times Circled	Rank	Factors
2	3	4	5	6	7		1		(Highest)	
	2	2	2	2	2		2		6	
	3	4	5	6	7		3		5	
		3	3	3	3		4		4	
		4	5	6	7		5		3	
			4	4	4		6		2	
			5	6	7		7		1	
				5	5				0	(Lowest)
				6	7					
					6					
					7					

3. In each pair of rows in the matrix, circle that factor number which is the more important of the two under consideration.
4. After completing evaluation of all pairs, count the number of times each number has been circled in the entire matrix,
5. Arrange the factor numbers in the "Rank" table according to the number of times circled. Note that it is quite possible for two or more factors to be circled the same number of times. This merely indicates that such factors have equal weight in the subsequent evaluation. Some ranks will remain empty if this occurs, indicating a jump in the order of importance of certain factors.

We have now established the relative importance of the various factors in a reasonably objective manner by systematically comparing each factor with each other factor on a mutually exclusive basis. The actual development of the emphasis curve for this report will be shown as an example of the technique.

I. CRITERIA FOR SYSTEM COMPARISON

The following criteria were considered to be significant in the comparative evaluation of the various systems described in this report. In accordance with the above instructions for development of the emphasis curve, they are not listed in any particular order other than that in which they occurred to the writer.

1. Weight

The objective is to provide a high degree of system optimization by providing a minimum weight unit consistent with economical manufacturing practices, high strength construction materials, and to meet design objectives and requirements.

2. Efficiency of Energy Absorption

A highly efficient energy absorption system reduces unit stroke resulting in an overall weight reduction.

3. Cost

This is largely self-explanatory. Small variations in cost should not be too important, but if one system were several times the cost of another, this would become significant.

4. Reliability

This is a function of the number of elements in the system, their individual reliabilities, and the effect of failure on system operation and personnel hazard.

5. Unit Life

The unit must operate a minimum of 1,000 cycles with 100,000 cycles desirable.

6. Maintainability

The unit should require a minimum of servicing with materials readily available to operating personnel.

7. Range of Adaptation

The unit should be readily adaptable to a wide range of high performance airplanes.

J. DEVELOPMENT OF THE EMPHASIS CURVE

These seven factors were used as the basis for the matrix described above. The matrix worked out as follows:

1 ① 1 ① ① 1	Factor	Times	Rank	Factors	
② 3 ④ 5 6 ⑦	Number	Circled	(Highest)		
② 2 ② ② ②	1	3			
3 ④ 5 6 7	2	5	6	4	Reliability
3 ③ 3 3	3	1	5	2	Efficiency
④ 5 ⑥ ⑦	4	6	4	7	Adaptation
④ ④ ④	5	0	3	1	Weight
5 6 7	6	2	2	6	Maintain-
5 5	7	4	1	3	ability
⑥ ⑦			0	5	Cost
6			(Lowest)		Unit Life
⑦					

The results of this evaluation indicate the following order of importance for the seven factors. By coincidence, there are no two factors of equal importance, although this has no specific meaning in itself.

Factor	Rank	Percentage of Total Weight
Reliability	6	29
Efficiency	5	24
Adaptation	4	19
Weight	3	14
Maintainability	2	9
Cost	1	5
Unit Life	0	0

Since the numerical total of the weighting factors is 21, emphasis curve assigns the above percentage values to the importance of each factor. A subjective examination of the relative weights discloses, for example, that reliability is twice as important as weight. It must be remembered that the relative weighting was originally established by the number of times that the given factor was considered more important than some other given factor, so the magnitude of the rank does have a significance in the analysis.

K. COMPARATIVE ANALYSIS OF VARIOUS SYSTEMS

Having established a weighting system for the various criteria of interest, the next step was to evaluate each of the systems and grade it for its effectiveness in meeting each of the criteria. A five-point grading system was used, since it was felt that evaluation to a greater accuracy than a 20% band was not possible at this early stage. Accordingly, each system was given a grade from one to five, with a score of five being given to that system or systems which best met each of the criteria, and a score of 1 to the worst. It will be noted that since "Unit Life" rated zero on the emphasis curve, it did not enter into the evaluation.

The results of this evaluation are shown on Table I.

L. CONCEPTS SUBMITTED

The appendix shows the concepts submitted in pictorial form. The four (4) most promising concepts, as selected at an evaluation meeting with the FDFM project engineer, are listed below:

1. Figure No. 16 - As depicted.
2. Figure No. 29 - As depicted.
3. Figure No. 29 - Modified by incorporating the control device of Figure No. 21.
4. Figure No. 34 - Modified by incorporating the control device of Figure No. 18.

TABLE I. COMPARATIVE ANALYSIS

Fig. No.	FACTORS/WEIGHT						Total
	Cost /1	Maint. /2	Weight /3	Adapt. /4	Efficiency /5	Reliability /6	
11	3 3	3 6	4 12	5 20	4 20	4 24	85
12	4 4	3 6	3 9	2 8	4 20	3 18	65
13	4 4	3 6	3 9	2 8	4 20	3 18	65
14	4 4	3 6	2 6	2 8	4 20	3 18	62
15	4 4	3 6	4 12	5 20	4 20	4 24	85
16	4 4	4 8	5 15	5 20	5 25	4 24	96
17	3 3	3 6	2 6	2 8	3 15	3 18	56
18	Wear Compensator - Could apply to all						
19	3 3	3 6	3 9	4 16	5 25	4 24	83
20	3 3	4 8	3 9	2 8	4 20	3 18	69
21	3 3	4 8	3 9	2 8	4 20	3 18	69
22	3 3	3 6	4 12	5 20	4 20	3 18	79
23	4 4	4 8	4 12	4 16	4 20	3 18	78
24	4 4	4 8	2 6	2 8	3 15	3 18	59
25	5 5	5 10	3 9	1 4	3 15	3 18	61
26	4 4	4 8	3 9	3 12	4 20	4 24	77
27	4 4	4 8	3 9	3 12	4 20	4 24	77
28	4 4	4 8	3 9	4 16	5 25	4 24	86
29	4 4	4 8	4 12	4 16	5 25	4 24	89
30	3 3	3 6	3 9	3 12	3 15	3 18	63
31	3 3	3 6	3 9	3 12	3 15	3 18	63
32	3 3	3 6	3 9	4 16	4 20	3 18	72
33	3 3	3 6	4 12	3 12	4 20	3 18	71
34	3 3	3 6	4 12	5 20	4 20	4 24	85
35	3 3	3 6	5 15	5 20	5 25	4 24	93

SECTION III

LITERATURE AND MATERIAL SURVEY

A. Literature Survey

Concurrent with Phase I of the project, to establish as many friction devices as possible, a literature survey was conducted to ascertain whether any of the methods developed had been previously utilized and to preclude any duplication of effort.

Initially a library search was conducted of friction technology and application. A review of various technical publications indicated a good deal has been written on the elimination of friction and wear from mechanisms and devices, but very little is available on the application of friction to dissipate energy, or to overcoming the effects introduced by the difference in static and dynamic coefficient of friction.

Secondly, a request was made to the DEFENSE DOCUMENTATION CENTER, Alexandria, Virginia, for any reports that might describe high temperature, friction-type, aircraft shock absorbers. A negative reply was received from this source of information. Alluding to the thoroughness of their search, some information was obtained on aircraft shock absorbers, which they hoped would lead to additional information on friction landing gear.

A third source of survey information on friction and materials was the DEFENSE METALS INFORMATION CENTER, Battelle Memorial Institute, Columbus, Ohio. DMIC Memorandum 185 - "Classification of DMIC Reports and Memoranda by Major Subject" was very helpful in locating high-temperature bearing materials and metals for elevated temperature use.

Patent Counsel at Cleveland Pneumatic assisted in the survey by preparing a state-of-the-art search for friction type shock absorbers, although a review of the information obtained was found to be of little interest to the project.

Other sources searched during the survey were:

1. Scientific and Technical Aerospace Reports Indexes - NASA.
2. Technical Abstract Bulletins - DDC.
3. U. S. Government Research and Development Reports.
4. Battelle Technical Review.
5. International Aerospace Abstracts Index.

Although much information was found on energy absorption devices for aero-space vehicles, very little has been written on systems in which heat is generated and energy dissipated by friction.

It can only be concluded that the approach is unique and warrants further development.

B. Friction Material Survey

Friction material manufacturers were surveyed for their applicability to the various designs early in the project to obtain detail design data of various friction materials for evaluation and selection.

Some of the features and design requirements that were requested are listed below:

1. Nearly constant coefficient of friction from -20°F to 1800°F (ambient temperature 1000°F).
2. Nearly equal static and dynamic coefficients of friction.
3. Little change in the friction coefficients as wear progresses.
4. Allowable face loadings and expected range of the coefficient of friction during the various modes of operation.

Seven (7) manufacturers were surveyed and favorable replies received from two (2) manufacturers. Of these two, it was eventually determined that the requirements could probably be met with the disc type of proposal and that a linear "shoe-type" of design was not readily adaptable to present equipment and fabricating techniques.

It can be assumed, therefore, that any concept requiring a "shoe-type" linear device would require additional development of friction materials and/or a compromise in the design requirements or parameters.

Present day copper or iron based sintered materials could very well handle temperatures of 1000°F and face pressures of 1000 psi, with little change in coefficient of friction when applied to the multiple disc type of design.

C. Materials for Elevated-Temperature Use

In the selection of materials for elevated-temperature use, a number of problems encounter the designer. Fabrication and economics, mechanical properties, oxidation and corrosion resistance and physical properties are some of the more important considerations when selecting a high temperature material.

At present, soak temperatures of 1800° F and maximum operating temperatures of 600° F to 800° F for landing gears could be expected in future high performance aircraft. Since the application for these high temperature materials will be in high performance aircraft where weight is important, the strength-to-weight ratio of the material is more important than strength. Ultimate Strength, Yield Strength, Creep Resistance, Notch Sensitivity, Ductility and Fatigue Strength are other mechanical properties that must also be considered.

Steels, Titaniums and super alloys are being considered in research programs now in progress or about to be initiated. Unfortunately, the technology for producing, machining, forming and joining these new alloys is not complete and any selection must be tempered by what is available at the inception of design effort. Undoubtedly, initial efforts will result in compromises to weight and the number of cycles of landing gear operation.

Vacuum arc remelt H-11 steel would probably be chosen for major landing gear components because of its ability to maintain its strength after repeated exposure to the expected operating temperatures. A limited use of Titanium alloys could be expected in secondary structures such as Torque Arms. Most bushings could be made of corrosion resistant steel and upper and lower bearings of beryllium-copper would be utilized.

Finally, a most thorough examination of thermal expansion coefficients must be maintained in order to assure complete reliability at elevated-temperatures. Because of the use of welded diaphragm metal bellows, and subsequent elimination of static and dynamic seals, larger tolerances could be applied to high temperature bearings and sliding applications.

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SECTION IV

PRELIMINARY DESIGN

A. PERFORMANCE REQUIREMENTS

Phase II of the project was initiated by the analysis and selection of an actual high performance aircraft on which to incorporate preliminary sketches of the selected landing gear concepts. Final selection of the U.S.A.F. F-100 aircraft main landing gear was made based on the technical data available at Cleveland Pneumatic because of the large quantities of F-100 gear that have been produced at Cleveland Pneumatic.

Complete drop testing information, stress analysis, technical manual, detail drawings and other landing gear data were readily available on the F-100 for use in preparing the preliminary drawings of the selected concepts. Some of the basic landing gear performance requirements for the F-100 are listed below:

PERFORMANCE REQUIREMENTS

1. Take-off Weight	37,386 lbs.
2. Landing Weight	22,050 lbs.
3. Load Factor	3.0 limit
4. Sinking Speed	10 Ft/Sec
5. Piston O.D.	4.25 inches
6. Cylinder I.D.	4.75 inches
7. Tire	30 x 8.8 type VII 22 P.R.
8. Wheel Speed	175 knots
9. Tire Pressure	275 psi
10. Stroke	10.135 inches

B. TECHNICAL DESCRIPTION

The landing gear shock strut for the F-100 aircraft is a typical cantilever single wheel, air-oil unit. Attached to the upper end of the outer cylinder are the trunnions that mount into the trunnion supports in the wing

structure. (See Figure 1). Also attachment points for the retraction actuator and side brace are located on the outer cylinder. Conventional torque arm links are mounted to the lower cylinder and piston lugs to prevent cylinder rotation.

A single, high pressure tire is mounted aft and outboard of the strut centerline. The gear proper pivots on the trunnion support fitting which is located on the wing rear spar. The side brace extends outboard and aft of the strut centerline and in a plane perpendicular to the trunnion axis to a fitting on the wing rear spar.

C. DESIGN DEVELOPMENT

1. Metered Linear Control

The original concept of Figure 15 depicted segmented brake shoes attached to the piston in a telescoping piston and cylinder arrangement. The shoes were expanded outwardly to engage the inside diameter of the cylinder. An "on" and "off" type control of the friction force was provided.

The linear damper concept was initiated because the motion we are trying to stop is linear and a linear type of damper would eliminate the need for changing linear motion into rotary motion and would also be quite similar to conventional brake designs.

For landing, the control arm at the top of the assembly is positioned so that the cam to which the control arm is attached is fully expanding the brake shoes. If the shoes were fully expanded for taxiing, telescoping action could be obtained only when the taxiing load reached the same magnitude as the landing load. Even then, the assembly would compress, but would not extend. Therefore, immediately after landing the control arm is moved so that the cam is rotated to present the low side of the cams to the cam followers. This unloads the brake shoes to enable the airplane to taxi on the bellows without the high friction. To compensate for wear and differential thermal coefficient of expansion, springs are shown between the cam followers and the brake shoes.

Initial design efforts indicated that large radial loads were necessary to generate the reactive loads in the friction material and the shortcomings of cam rollers was soon apparent. Preliminary designs were therefore directed toward the use of circulating balls (see Figure 1) which were actuated by a metered control rod. The control rod is tapered at the lower end to allow for load build-up and elimination of the effects due to the differences between the static and dynamic coefficient of friction.

Essentially a non-regenerative constant friction device resulted

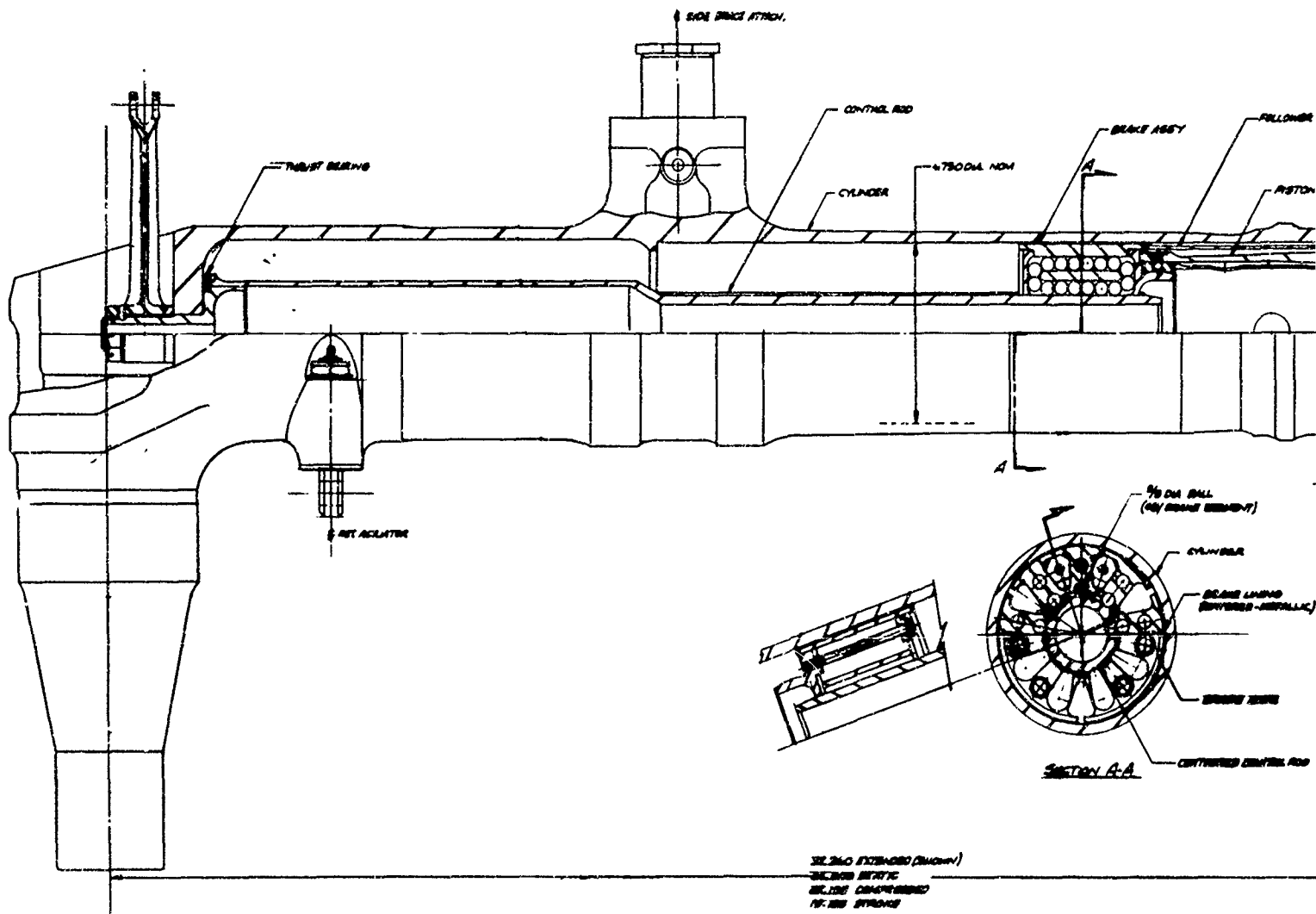
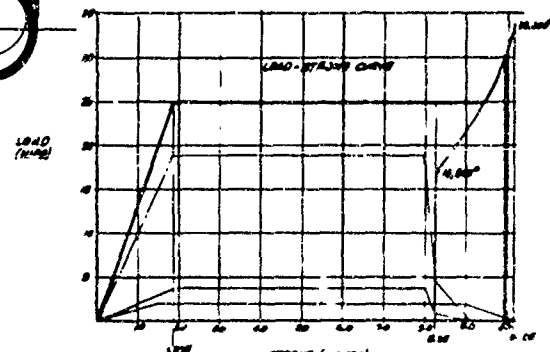
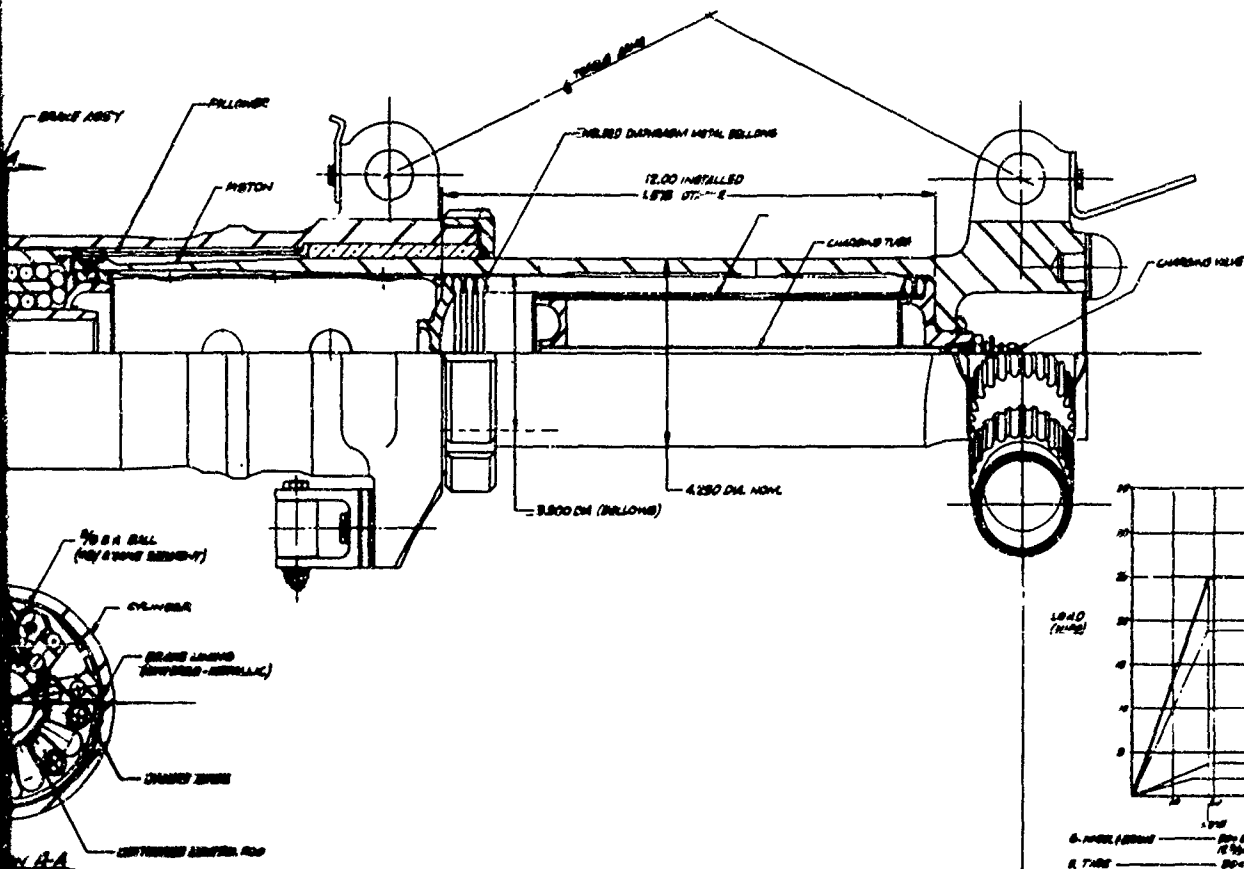


Figure 1. Drawing SK 1552

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1. PISTON ARMY — 20.00 TYPE
 2. PISTON — 10.00 S-2 TYPE ARMY
 3. PISTON — 10.00 S-2 TYPE ARMY
 4. LANDING HEIGHT — 20.00 LB.
 5. PISTON HEIGHT — 20.00 LB.
 6. PISTON TYPE — 10.00 S-2 TYPE ARMY
 7. PISTON TYPE — 10.00 S-2 TYPE ARMY
 8. PISTON TYPE — 10.00 S-2 TYPE ARMY
 9. PISTON TYPE — 10.00 S-2 TYPE ARMY
 10. PISTON TYPE — 10.00 S-2 TYPE ARMY

DATE: 10/10/50

that dissipates the total kinetic energy through lower bearing-piston friction, brake shoe friction, ball (rolling contact) friction, tire energy absorption and energy absorption through actuation of the metal bellows.

By careful control of the metering rod configuration and pre-determined tapers it is expected that strut efficiencies of 80 to 85 percent can be developed. The brake segments are pivoted at a spherical joint attached to the piston, thus eliminating binding problems associated with drag loads that result from tire spin-up and spring-back. This is essential in all designs in order to maintain complete concentricity between control rod, friction shoes and the cylinder I.D.

Experience indicates that if strut reaction factors equal to a vertical acceleration of 2 g during taxiing are selected, then adequate ride control can be expected and approximately 1.875 inches of bellows stroke is necessary. This necessitated a minor change in static position from that of the F-100 aircraft geometry which was a design compromise in keeping with accepted design practice and also which facilitated the design of the welded metal diaphragm bellows.

Because complete drop testing data was available for the F-100 gear, accurate drag load results were available for calculating bearing reactions and strut friction. Also, tire deflections and impact energies contributed toward the accuracy of preliminary design calculations. Energy levels, radial loads and resultant face loadings on the segmented shoes were relatively high, although the time under load is extremely short (0.20 seconds) which results in horsepower per square inch of lining material ratings of approximately $2.0 \frac{\text{HP}}{\text{IN}^2}$ which is typical when comparing lining performance between one brake and another.

Obviously any linear friction shock absorber using a shoe type design will suffer from the vagaries of bad fits, hot spots, non-uniform wear, chatter and other brake problems that have plagued the manufacturers and would require further research and development work to overcome.

The descent velocity of the proposed gear is estimated to be in the range of 3 ft/sec to a maximum of 10 ft/sec. In the friction type landing gear where no metering device exists to vary pressure, some sort of sensing device must be devised to enable the gear to provide varying amounts of pressure induced on the brake pads corresponding to rate of descent thereby allowing the gear to cycle completely regardless of load imposed.

The use of an accelerometer to sense the rate of sink coupled with a rotary solenoid (proportional) would allow the rate of sink to vary over a wide range. Input to the solenoid would rotate a control rod to the correct point, to insure the necessary pressure to absorb the landing load,

and cycle the gear through the landing phase into the taxi phase without interruption or lock-up due to overpressure on the cylinder in a light load landing. This system would allow smooth operation regardless of approach of the ground by the aircraft. A mechanical linkage as required on Figure 1 could replace the rotary solenoid in the system but a proportional input device is still required to control the amount of turn exerted on the control rod.

A transducer could also be used to indicate the change in pressure induced by a varying sinking speed. The output EMF could be used to generate a signal which would operate a proportional rotary solenoid actuating the control arm. Both systems would require an interlock to prevent the control arm from overriding and exerting too much or too little pressure.

Pneumatic devices to indicate pressure change on landing have been virtually eliminated due to the temperature range expected on this type system. Since the initial pressure on the system could fluctuate substantially due to temperature, it would make the initial reference point in the system unstable and hence the system would therefore be as erratic as the initial point of reference. Temperature also rules out the use of hydraulic indicators in the system.

A magnetic control device could be incorporated into the system but certain drawbacks make it seem less desirable than the accelerometer or transducer. An iron rod attached to the cylinder could be suspended in a coil attached to the piston. As the rod travels through the coil, a magnetic field is established the intensity of which is correlated to the speed of the rod traveling through the coil. This output could be transmitted to a rotary solenoid or electro-pneumatic actuator which would rotate the control rod as required. This system could be hampered considerably by bending in the system due to drag loads which would bring the rod and coil in contact and defeat the system. An insulated coil would solve part of this problem but the system would still be exposed and subject to damage from handling, gear maintenance and flying objects during taxi, landing and take offs. This system and the systems employing the accelerometer and transducer all rely on electrical connections which could fail the system should a wire break or shake loose. Therefore an overriding fail safe mechanical system which by necessity would not be as exotic, would insure safe troublefree landings.

2. Velocity Sensitive Control

Another example of design development after concept selection is to be noted in the development of figure number 34. As the preliminary design effort progressed it became apparent that the ideal design solution for a friction shock absorber must embody the following principles:

- 1.) Absorb landing energy with the brake device set at a maximum load rate.
- 2.) A self-energizing device is desired to assist in loading the brake.
- 3.) The mechanism must be pre-set at the minimum static reaction of the shock strut in order that the unit will stroke to engage the taxi bellows.
- 4.) The unit must disengage at the end of its travel to extend freely.
- 5.) The unit must re-engage in the extended position to be ready for the next cycle.

Figure 34 employed a long lead screw and nut arrangement wherein the axial load is transmitted from the screw to the other parts in the following order: Nut, rotating discs, stationary discs, bellville springs and housing.

The brake is self-energizing since the axial load applies the braking disc force. However, the bellville is adjusted to cause an initial load of slightly less than the minimum static reaction of the gear and the vertical travel of the nut is limited so that the design load cannot be exceeded. Another set of bellville springs is used to provide a taxi spring when the screw is not fully compressed. To insure extension, a helical spring is shown inside the upper end of the screw to produce an extension force.

In order to eliminate as many extraneous systems from the unit as possible and to employ a completely self-contained mechanical device to absorb the landing loads, a unit which would use nearly full stroke, with the force developed being essentially constant, at a magnitude just large enough to dissipate all of the energy put into the shock absorber is desired. The vertical load would then be proportional to the square of the sinking speed.

A method was thereby evolved using a centrifugal type of governor to adjust the maximum load to the sinking speed. By incorporating such a control device high loads are developed only at high sinking speeds and low loads at low sinking speeds. Since low sinking speeds are far more numerous, much is gained by adjusting the load to the sinking speed rather than accepting the maximum sink speed load value for all landing velocities.

Figure 2 depicts a multiple disc type of brake unit with rotating and stationary discs. The unit is driven by an overhauling acme screw with a quintruple thread which is ball-ended to run concentric in the cylinder and eliminate strut deflection affects in the unit. Linear forces are then transferred to the rotating nut which in turn transmits the angular velocity

through a sprag one-way clutch. The clutch drives the governor weights and the rotors of the brake assembly.

Centrifugal force causes the weighted arms to change their plane of rotation and thus apply braking force through the lever arrangement shown. The load is applied to the disc through a bellville spring. The bellville spring washer, since it has nearly a zero rate, or a practically horizontal load-deflection curve at the flat position, causes a more nearly constant load for short motions than that which would otherwise be attained if the axial load were applied directly to the discs. Without the springs, the load would be proportional to the square of the velocity. They also permit with spacers, setting of a minimum friction load.

In addition to the initial bellville adjustment, a sprag clutch was incorporated for automatically adjusting for wear of the brake discs thereby eliminating the need for constant adjustment. It is conceivable that mechanical devices could also be installed to indicate replacement of the discs as wear progressed and thus reduce maintenance to a minimum.

The overhauling screw contacts a welded diaphragm metal toroidal bellows with a torous contour at the static position to provide for support of the aircraft and an airspring for taxiing while on the ground. The use of a power screw is one of the standard methods for transmitting high loads in machines and also one which develops considerable friction. Efficiency, wear, and heating then become prime considerations in their design.

If the helix angle is sufficiently great or the coefficient of friction small, the axial load may be sufficient to turn the nut, and the screw is said to "overhaul". In the limiting condition, no torque is required to lower the load and the axial load just balances the friction forces.

Calculations associated with the design of an overhauling acme screw are shown in the appendix.

Total energy requirements for the F-100 main landing gear are 273,000 in lb. of which approximately 38,000 in-lb. of energy is absorbed by the tire leaving 235,000 in lb. to be absorbed in a 10.0 inch stroke. Assuming that the strut friction unit is 90 percent efficient the strut load is:

$$(1) \quad F = \frac{235,000}{0.90 (10)} = 26,000 \text{ lbs.}$$

Assume also that the tire deflects first and this more gradual load build-up mitigates the effects of "static" coefficient of friction.

$$(2) \quad \text{Acceleration} = \frac{26,100 (32.2)}{14,600} = 58 \frac{\text{FT}}{\text{SEC}^2}$$

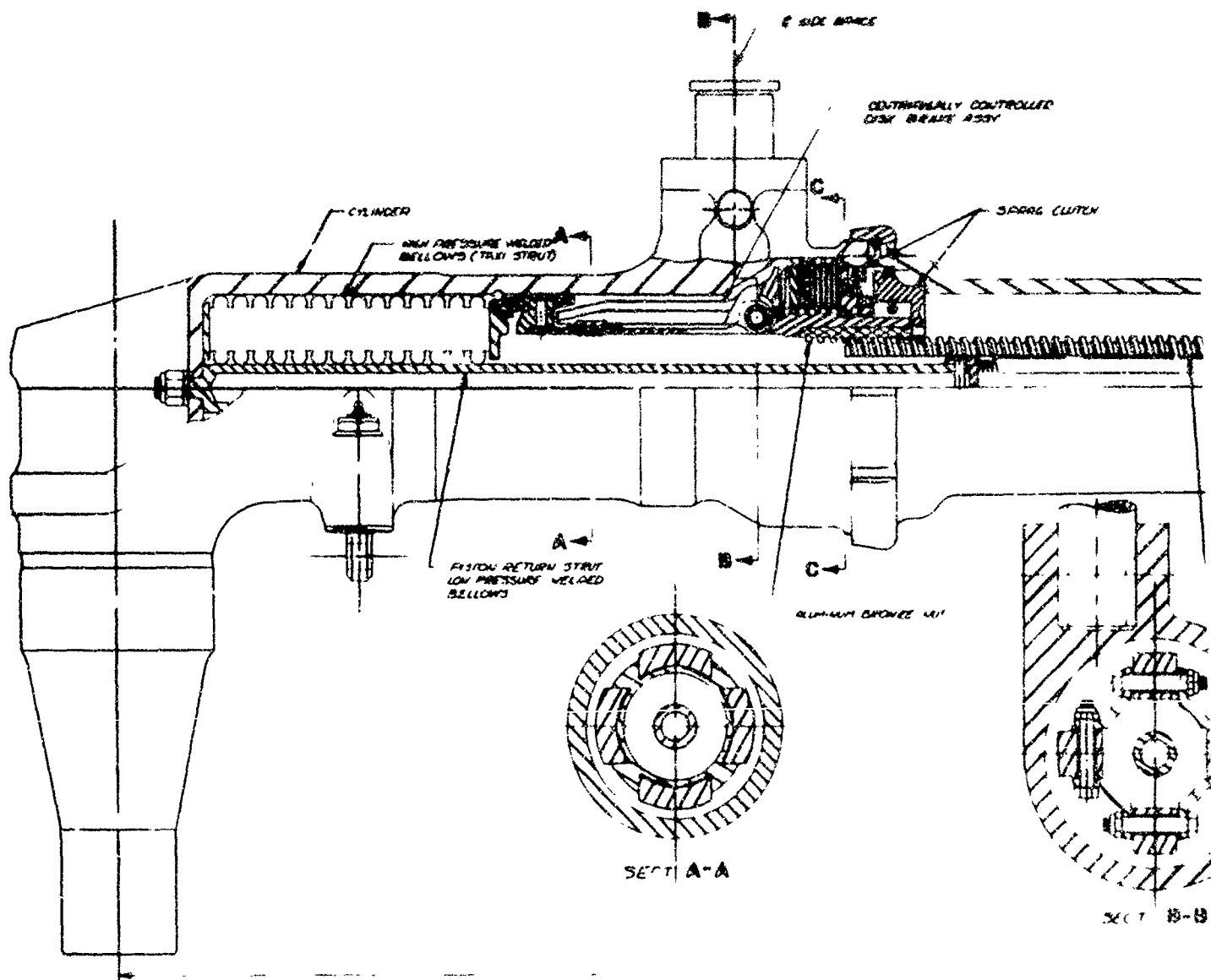
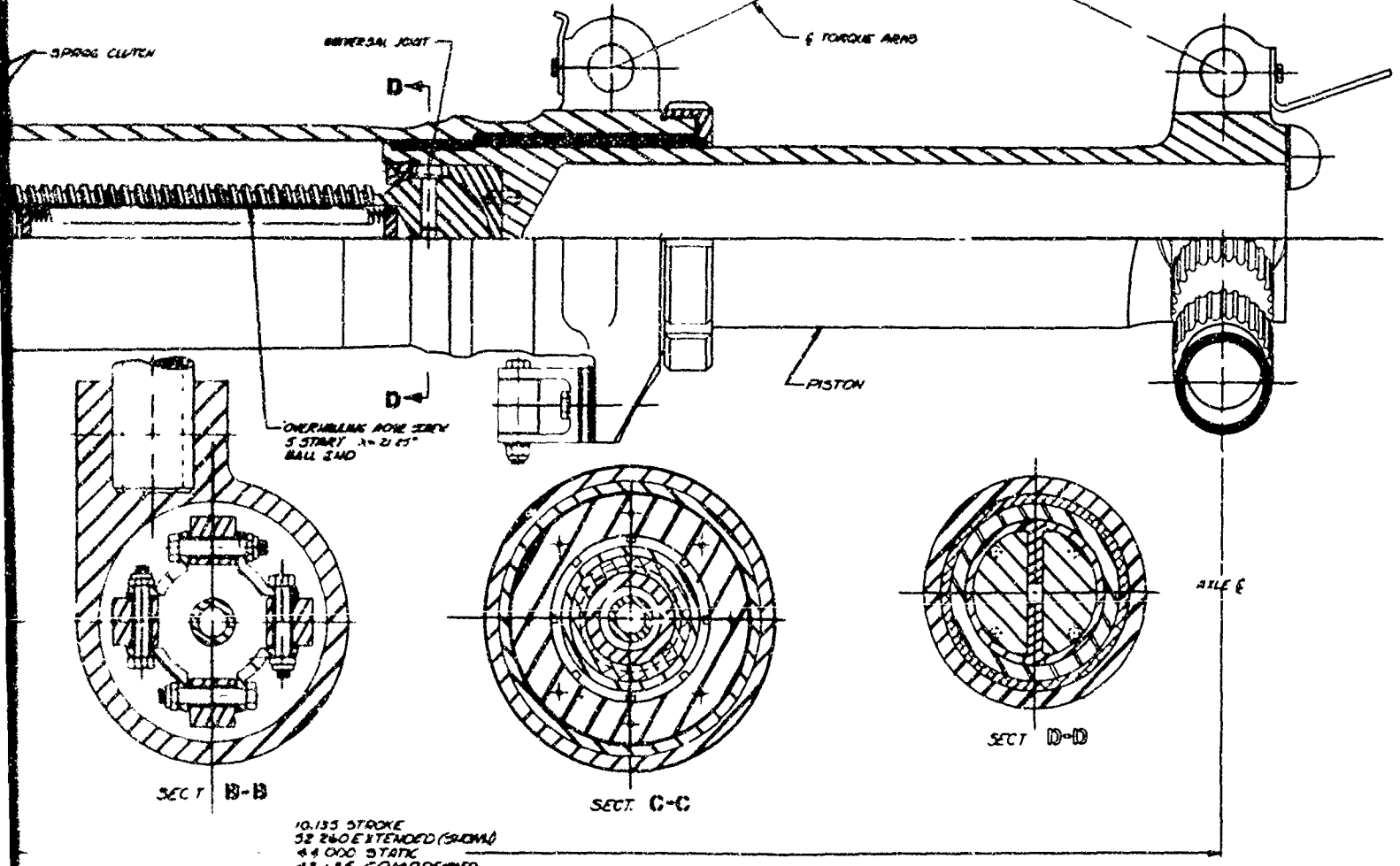


Figure 2. Drawing SK 1555

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CONTROLLED
PRESSURE



Acceleration versus stroke is plotted in Figure 3. Strut piston velocity versus stroke is plotted in Figure 4 and was prepared from drop test records developed on the F-100 aircraft landing gear.

(3) The ratio of nut rotation to linear stroke is:

$$\frac{2\pi \text{ RAD.}}{1.95 \text{ IN.}} = \frac{3.22 \text{ RAD.}}{\text{IN.}} = \frac{38.7 \text{ RAD.}}{\text{FT.}}$$

Angular velocity of the nut versus stroke is plotted in Figure 5 from the piston velocity versus stroke and the ratio of nut rotation to linear stroke. Finally centrifugal acceleration versus stroke is plotted in Figure 6.

The centrifugal governors are set for release at 9.0 inches of stroke and/or an acceleration of 1150 ft/sec². The force required at each governor for a unit having four weights is found to equal 177 pounds (see appendix) and the centroid and weight of the unit is calculated from the following:

$$\begin{aligned} (4) \quad P &= 1150 \text{ ft/sec}^2 (\text{weight}/32.2) (\bar{x}) \\ &= 35.7(W)(\bar{x}) \end{aligned}$$

Centroid and weight were found by trial and error in order to fit the envelope required. A unit weight of approximately 1.50 pounds was subsequently found to fit the required space and provide the necessary force at the brake unit.

D. BELLOWS DESIGN

Bellows design and development was instigated to overcome the problems associated with present seals at high temperatures, provide for high pressures to support the aircraft at static positions, thus eliminating heavy mechanical springs and/or friction release devices that would be complex and weight consuming and to provide for strut reliability.

A survey of the bellows industry was initiated and resulting technical discussions determined that for high pressure (e.g. 1500 psi at the static position) high temperature, and relatively large deflections, the state of the art of design and fabrication of bellows would require substantial advancement.

Development of welded diaphragm metal bellows from the analytical viewpoint is an extremely complicated one and beyond the scope of this report, from the designers viewpoint there are now being developed several basic diaphragm contours which can be used to establish reasonable envelopes and performance characteristics.

A torus diaphragm contour was selected because it has excellent

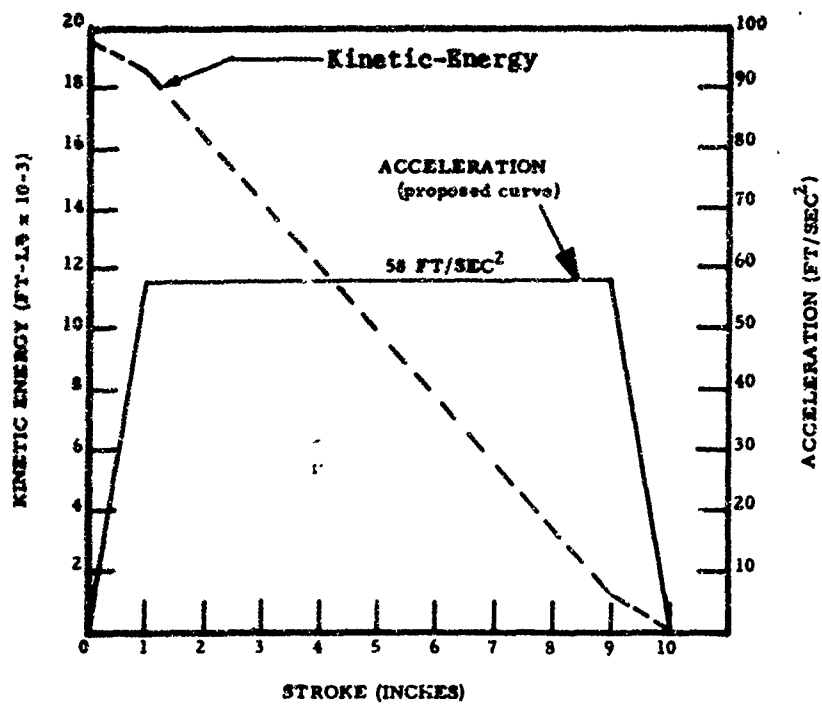


Figure 3. Acceleration vs Stroke

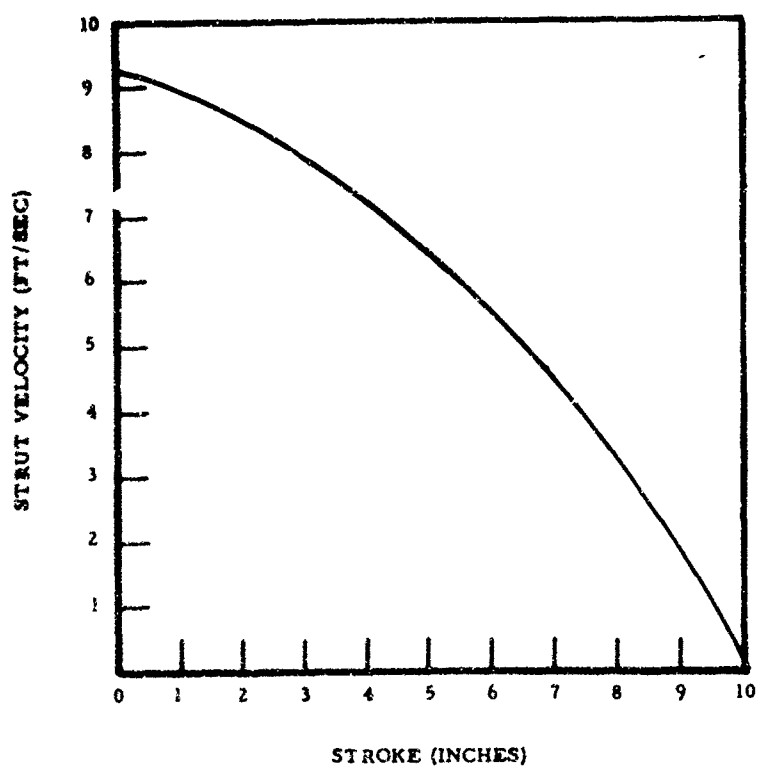


Figure 4. Piston Velocity vs Stroke

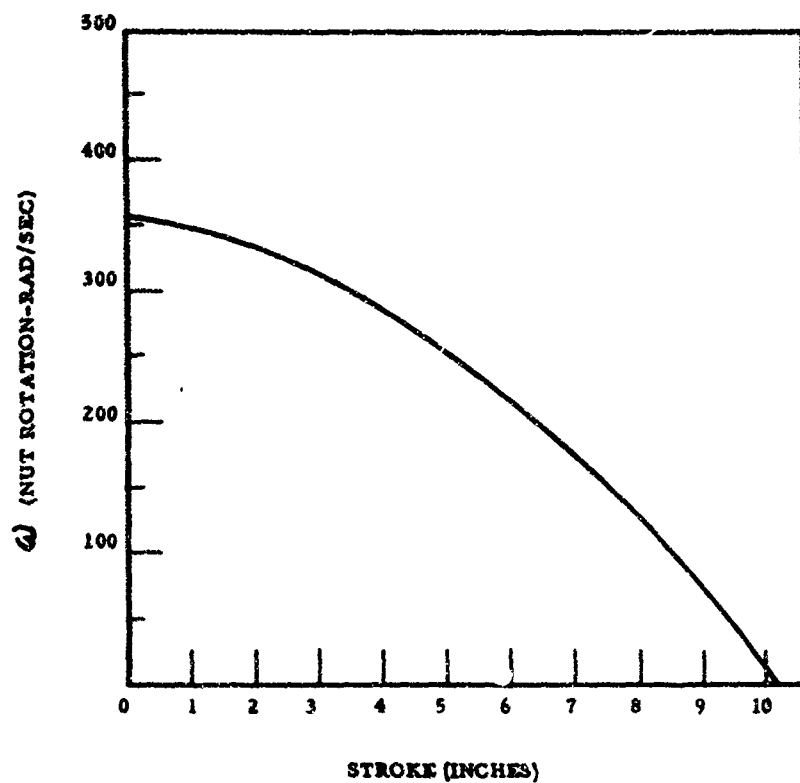


Figure 5. Angular Velocity vs Stroke

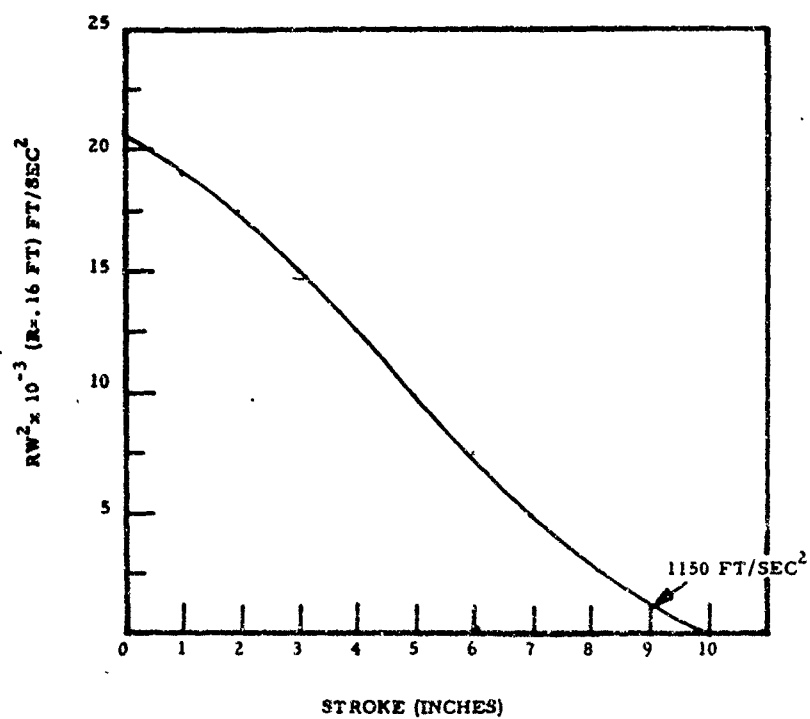


Figure 6. Centrifugal Acceleration vs Stroke

resistance to pressure, has fair long-stroke capability, has excellent linearity of force output with pressure, good linearity of stroke with pressure and high spring rate. Other performance requirements for bellows not related to contour are: temperature, in that it affects spring rate; strength of material or corrosion resistance; and life as a function of stress levels.

Because of the relative stiffness of the torus diaphragm contour, the stroke per convolution was low and resulted in a free length of approximately 12.0 inches to obtain 1.875 inches of stroke. This in itself, under normal application would tend to increase the length of shock struts when installed in the upper cylinder. Normally, it would be well to take advantage of the space available in the lower piston I.D. while sacrificing some decrease in bellows effective area.

Optimum bellows development would provide for a maximum stroke in a minimum installed length, a bellows torus section pressurized externally to increase the life of the unit and specialized charging systems to limit pressure increases due to high temperature environment. A general analysis would indicate that two systems are required. The first, or pressure relief system, would be energized upon gear retraction; whereupon an increase in pressure due to temperature, would discharge the nitrogen through a relief valve. Thus, a preset maximum pressure would be maintained. The second, or charging system, would be energized on gear extension and subsequent gear cooling in the air stream. A charging valve would be automatically opened to the system to recharge to a predetermined minimum pressure as the temperature drops and pressure is required. Separate systems are necessary to prevent pressure relief during taxi operations and subsequent pressure increase.

Generally, the above comments applied to all friction strut concepts, except with minor design variations to the bellows location, and the use of toroid section contours in certain specific applications.

The "Constant Friction Shock Absorber" depicted in Figure 29 and modified by the control device of Figure 21 were the remaining concepts developed into preliminary designs. Both devices were linear type units that required the same radial loads to obtain the necessary strut loads and energy dissipation as noted in the development of Figure 16. Figure 29 was basically a modification of Figure 26 and 27. It differed in performance in that it produces constant friction regardless of the direction of stroking. The use of tapers presented problems of overcoming friction in the tapers to allow for lower spring loads, as the unit frictional force is related only to the spring force. Needle rollers were incorporated into the final design along with bellville springs to provide the high loads necessary.

The control device of Figure 21 was incorporated to provide a device which would relieve the friction on the extension stroke. Steel balls

are used between the brake actuating member and the brake shoe. In the brake actuating member are tapered recesses in which the balls are retained. Relative motion between the brake actuating member and the brake shoes in the direction of sliding motion of the braking surfaces is provided. The tapered recesses then jam the balls into the small space when the drag of the brake shoes moves the brake shoes in that direction during strut compression. The travel of the balls is limited by stops. So the brake is fully on in this case. When the strut tends to extend due to the force of the taxiing spring, the brake shoe moves with the extending member due to the friction between the two. These two members must move in unison until the balls reach their stops in the unjammed position, since the balls will roll freely with very little friction.

This device provides a one-way brake which develops high friction during the compression stroke and little friction during the extension stroke. An additional advantage of the device is that relative motion is established between the brake shoes and the member upon which they ride before the brake is fully applied. Thus, if the static coefficient of friction is greater than the sliding coefficient no detrimental characteristics are obtained.

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SECTION V

MODEL DEVELOPMENT

A. DESIGN PARAMETERS

Model development was based on the preliminary designs of the full scale landing gear and a study conducted to establish proper model scaling. Model development was also based on the following considerations:

- a.) Size such that the model can be tested in the WPAFB No. 1 drop test machine (minimum 500 pounds; maximum 3600 pounds).
- b.) A size that can be fabricated by utilizing readily available bar stock which requires a minimum amount of machining.
- c.) Materials to be used should be selected for their economical features, machinability and availability.
- d.) Temperature considerations: even though the units do not need to have superior temperature qualities, they should reasonably well simulate the physical characteristics that are envisioned to be used in final design.

B. DIMENSIONAL ANALYSIS

The dimensional system used in arriving at the dimensionless ratios is the force-length time, (F, L, T) system, with force designated as pound-force. Mass, then, is expressed as:

$$\frac{\text{Force}}{\text{Gravity Acceleration}} \quad \text{or force-length}^{-1} \text{ -time}^2 \text{ (FL}^{-1} \text{ T}^2\text{)}.$$

The primary feature of scale modeling is that any dimensionless ratio of the prototype is equal to the corresponding dimensionless ratio of the model. This is a basic and accepted, though not often used, mathematical operation.

The length ratio chosen was 2 to 1, as yielding a convenient size model, approximately 27 inches in length. Densities of materials were maintained in the ratio of 1.0 to 2 and moduli of materials at 1.0 to 1.0. With a length ratio of 2 to 1 a force ratio of 4 to 1 resulted, giving a force that could be developed in the WPAFB No. 1 drop test machine.

The following table lists the dimensionless ratios that were involved. The ratios were derived by dividing a prototype quantity by the corresponding model quantity; e.g., prototype mass by model mass. The upper case letters refer to prototype values, and the lower case to the model.

The second table includes the pertinent quantities selected, as well as quantities relating to the modeled concept.

TABLE II. DIMENSIONLESS RATIOS

1. Length	$L/1$	$= 2/1$
2. Gravity & Accelerations	$A/a = G/g$	$= 1/1$
3. Velocities	$V/v = \left(\frac{G}{g}\right)^{1/2} \left(\frac{L}{1}\right)^{1/2} = \left(\frac{1}{1}\right)^{1/2} \left(\frac{2}{1}\right)^{1/2}$	$= 1.414/1$
4. Time	$T/t = \left(\frac{L}{1}\right) \left(\frac{v}{V}\right) = \left(\frac{2}{1}\right) \left(\frac{1}{1.414}\right)$	$= 1.414/1$
5. Moduli $\frac{\text{Force}}{\text{Area}}$	$\frac{E}{e} = \left(\frac{M}{m}\right) \left(\frac{t}{T}\right)^2 \left(\frac{1}{L}\right)$	$= 1/1$
6. Mass & Weight	$M/m = \left(\frac{T}{t}\right)^2 \left(\frac{L}{1}\right) = \left(\frac{1.414}{1}\right)^2 \left(\frac{2}{1}\right)$	$= 4/1$
7. Energies	$\frac{KE}{ke} = \left(\frac{M}{m}\right) \left(\frac{t}{T}\right)^2 \left(\frac{L}{1}\right) = \left(\frac{4}{1}\right) \left(\frac{1}{1.414}\right) \left(\frac{2}{1}\right)$	$= 8/1$
8. Densities	$\frac{D}{d} = \left(\frac{M}{m}\right) \left(\frac{1}{L}\right)^3 = \left(\frac{4}{1}\right) \left(\frac{1}{2}\right)^3$	$= 1/2$
9. Forces	$\frac{F}{f} = \frac{MA + MG}{ma + mg} = \left(\frac{M}{m}\right) \left(\frac{A+G}{a+g}\right) = \frac{M}{m}$	$= 4/1$
10. Angular Velocity	$\frac{\Omega}{\omega} = \left(\frac{V}{v}\right) \left(\frac{1}{L}\right) = \left(\frac{1.414}{1}\right) \left(\frac{1}{2}\right)$	$= .707/1$
11. Angular Acceleration	$\frac{\alpha}{\alpha} = \left(\frac{V}{v}\right)^2 \left(\frac{1}{L}\right) = \left(\frac{1.414}{1}\right)^2 \left(\frac{1}{2}\right)$	$= 1/1$

TABLE III. PROTOTYPE-MODEL QUANTITIES

PARAMETER	PROTOTYPE	MODEL
1. Strut Stroke (in)	10.135	5.067
2. Drop Weight (lbs)	14,600	3650
3. Sink Speed (Ft/Sec)	10.0	7.072
4. Total Energy (in-lb)	273,000	34,125
5. Drop Height (in)	18.7	9.35
6. Wheel Speed (knots)	175.0	123.7
7. Acceleration (Ft/Sec ²)	58.0	58.0

C. TECHNICAL DISCUSSION

The final configuration for the scale model design is depicted in Figure 7. A universal mounting fixture was utilized in order to provide a simple attachment for all test units. Because of the basic configuration many components are interchangeable such as; torque arms, axle fitting, cylinder, and bearings. An 18 x 5.5 type VII - 8 ply rating, tubeless tire was selected because of its availability, adaptability, and because of its impact energy properties which closely approximated the required one-eighth energy capacities of the prototype tire (a 30 x 8.8 type VII - 22 ply rating).

A piston O.D. of 2.125 inches and a cylinder I.D. of 2.375 inches were maintained to provide the required one-half scale sizing. Loads were in the ratio of one to six, in that a load factor of two-thirds of the prototype and a load scale of one to four were used on the scale model. Each unit was stressed for a vertical load of 8700 pounds, ultimate and a drag load of 4400 pounds ultimate.

Generally, major components were designed to a stress level of 180,000 psi minimum tensile strength and are to be manufactured from 4340 steel. It is expected that although this material does not have superior temperature qualities, it will simulate the physical characteristics used in a prototype design. Although designed primarily as an energy absorption device the unit lends itself to complete drop testing, including tire spin-up and elevated temperature testing. By protecting or shielding the tire it is expected that the unit could operate at 800° F for limited drop testing.

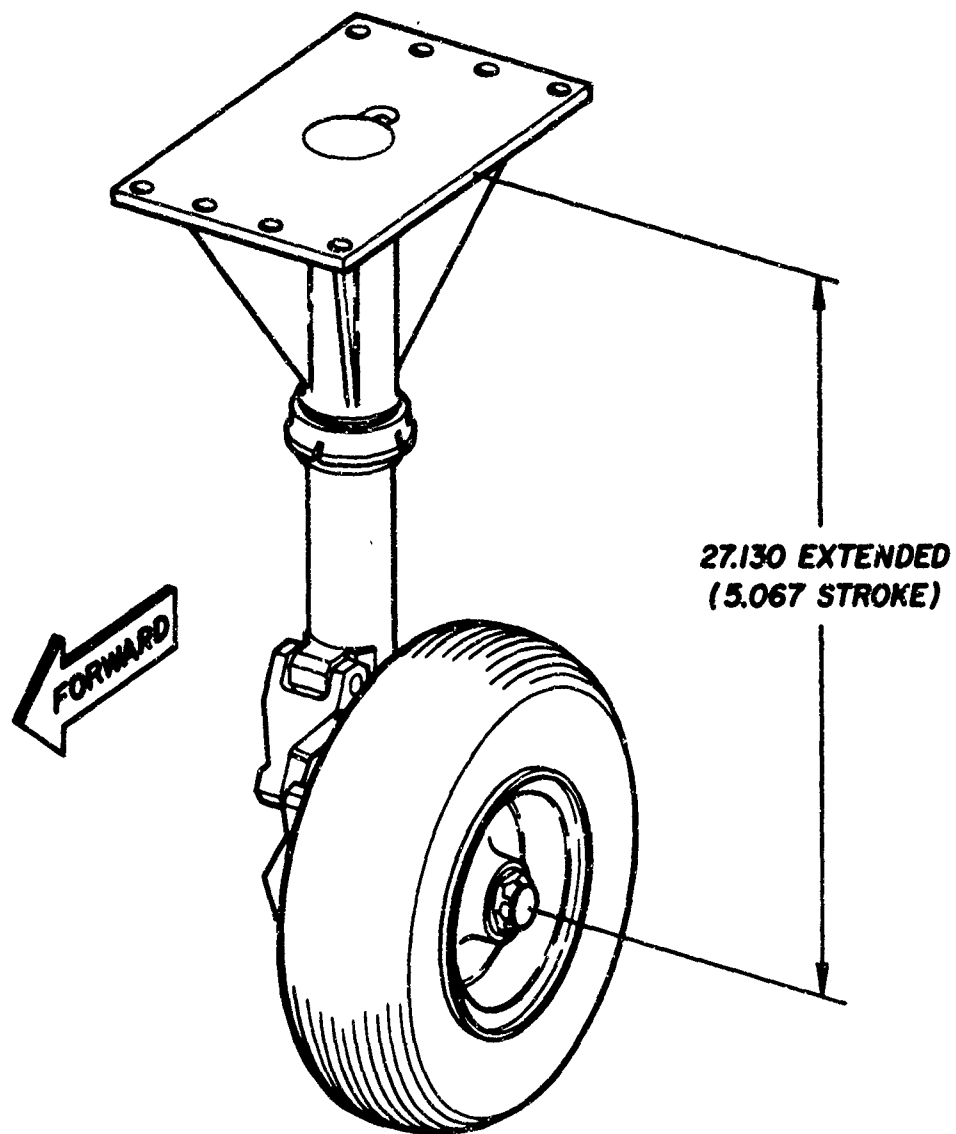


Figure 7. Friction Shock Absorber (Model Test Unit)

Figure 8 shows the centrifugally controlled disc brake unit as developed for the scale model. Load input through the piston causes the power screw to "overhaul" and transfer the linear forces to the rotating nut, through the sprag "one-way" clutch which drives the governor weights and the rotors of the brake assembly. An interesting aspect of the governor design is that since densities are in the ratio of one to two, it was necessary to select a material that was twice as heavy as steel to maintain the one-half size ratio for detail parts. Selection of a tungsten, copper and nickel alloy that weighs 0.67 pounds per cubic inch and is strong, ductile and machinable provided the answer. The adverse density ratio as developed proved to be the only inadequacy in the model scale design phase. Total model similiarity could be developed, but only at increased costs which are not actually required at this phase of the project.

In figure 8 the welded diaphragm metal bellows is shown at the top of the cylinder where the power screw causes the bellows to compress for approximately 0.875 inch at the end of its travel.

The metered linear control shock absorber is shown in Figure 9 as it has been adapted to the basic test unit configuration. The brake unit is energized when the circulating balls are engaged with the tapered control rod and disengaged when the control rod handle is rotated after linear motion has been arrested. The welded metal diaphragm bellows in this unit is installed in the lower piston where it is stroked on contact with the control rod when the static position is attained.

It is expected that manual control of the control rod handle should be sufficient during test operations of the model design. System sophistication can be realized once the prototype design is undertaken as described in Section IV "Design Development".

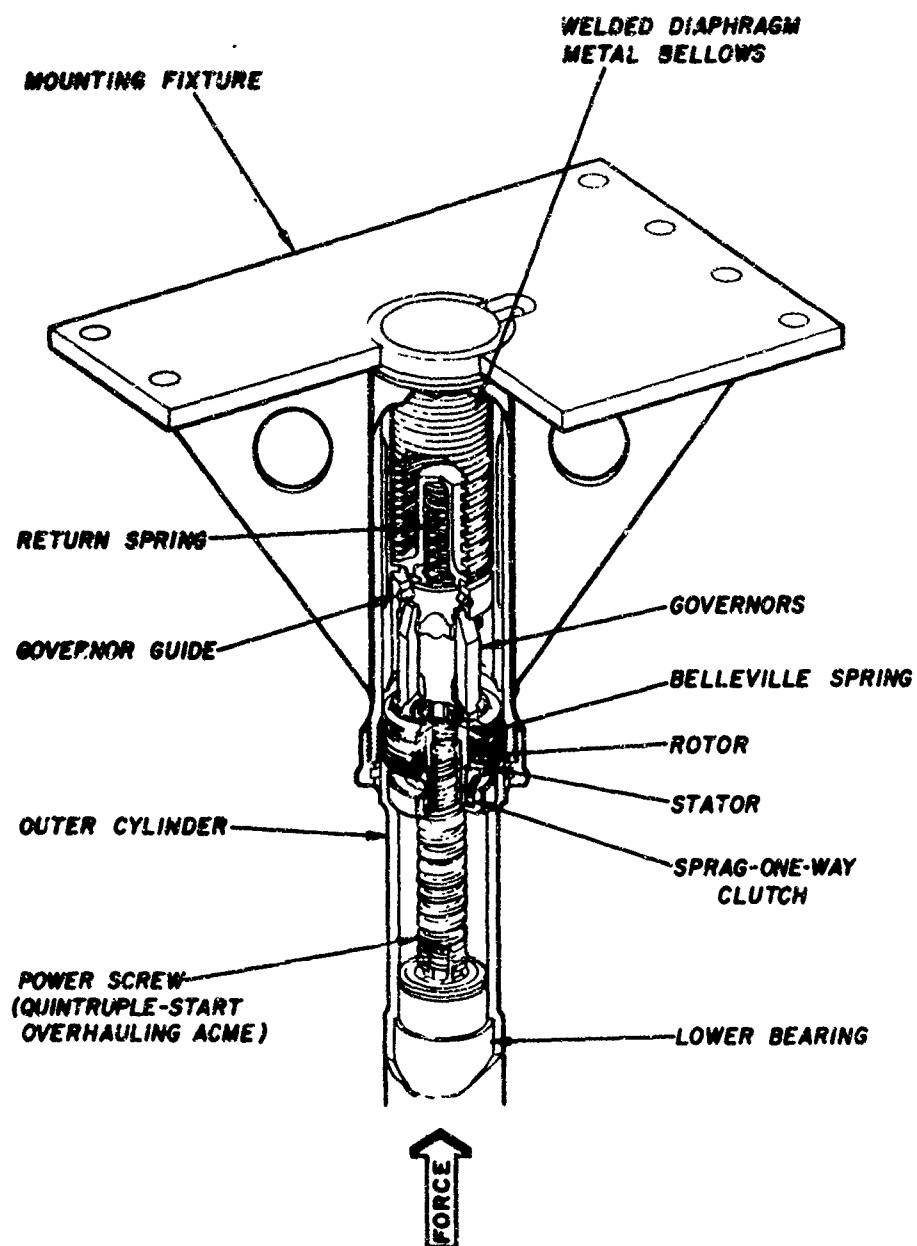


Figure 8. Friction Shock Absorber Centrifugally Controlled Disc Brake

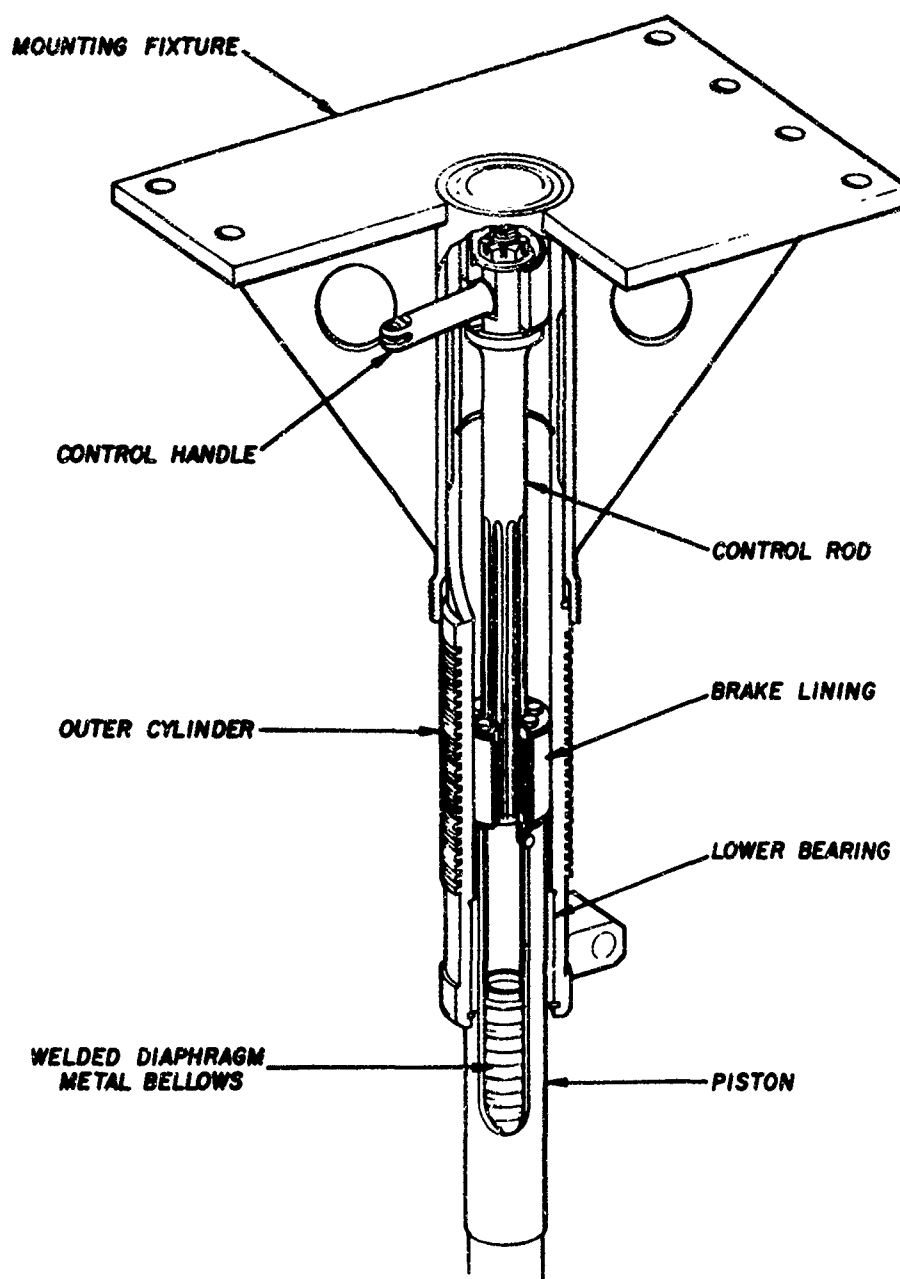


Figure 9. Friction Shock Absorber (Metered Linear Control)

SECTION VI

CONCLUSIONS

In the final analysis and evaluation of the selected concepts it is necessary not only to compare each unit with respect to one another but also with regard to cost, weight, reliability efficiency and future adaptation to various high performance aircraft.

An analysis of this sort indicates that the centrifugally controlled disc brake has greater advantages than the linear controlled friction devices. The rotary device is self-sufficient and requires no external systems excepting that required to maintain bellows pressure control. The disc type unit is also readily adaptable to present manufacturing facilities, techniques and material advances. The discs themselves allow for full contact area and are used over and over with little wear anticipated. It is also expected that as landing gears become larger and temperature problems increase, the friction type of unit will alleviate and overcome these difficulties.

An advantage of the rotary type is that a contracting shock absorber apparatus can very easily be adapted to the unit because of brake release mechanisms, etc., and the gear when retracted would have less length and thus be more easily stored in smaller wheel wells. Further, the non-regenerative unit contains no stored energy and would eliminate rebound problems, and therefore be very useful for rough-field operations.

Generally, constant friction devices would enable the use of lower load factors and reduce overall aircraft weight. Pneumatic retraction mechanisms would result in further aircraft weight reduction upon removal of hydraulic equipment and systems. Also, wheel well cooling equipment at the elevated temperatures would no longer be required.

Further testing and development is anticipated during model study which will establish physical properties and design data for prototype design.

Utilization of friction as the energy dissipator in aircraft shock absorbers is expected to advance the state-of-the-art and eliminate the necessity for inefficient material deformation units.

APPENDIX A

Pictorial Presentation of Concepts

This appendix contains the concepts as they were initially submitted during the course of the project. The concepts have not been arranged in any particular order and no attempt has been made to properly size the components but merely to portray the idea or thought associated with the use of such friction devices. Except in the case of those concepts selected by the FDFM project engineer for further study, no descriptions are given for the concepts in order to avoid a long report.

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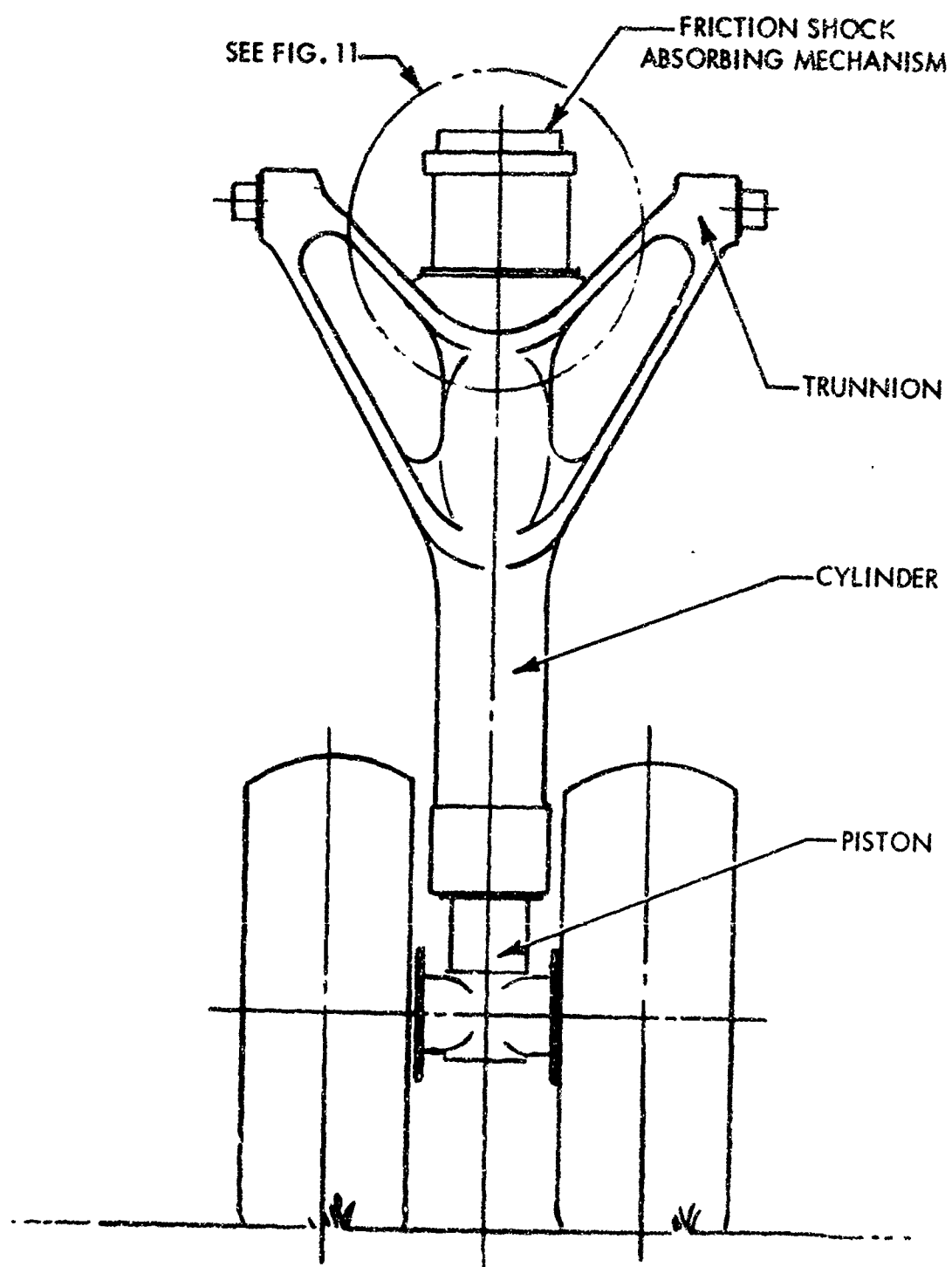


Figure 10. Cantilever Type Landing Gear

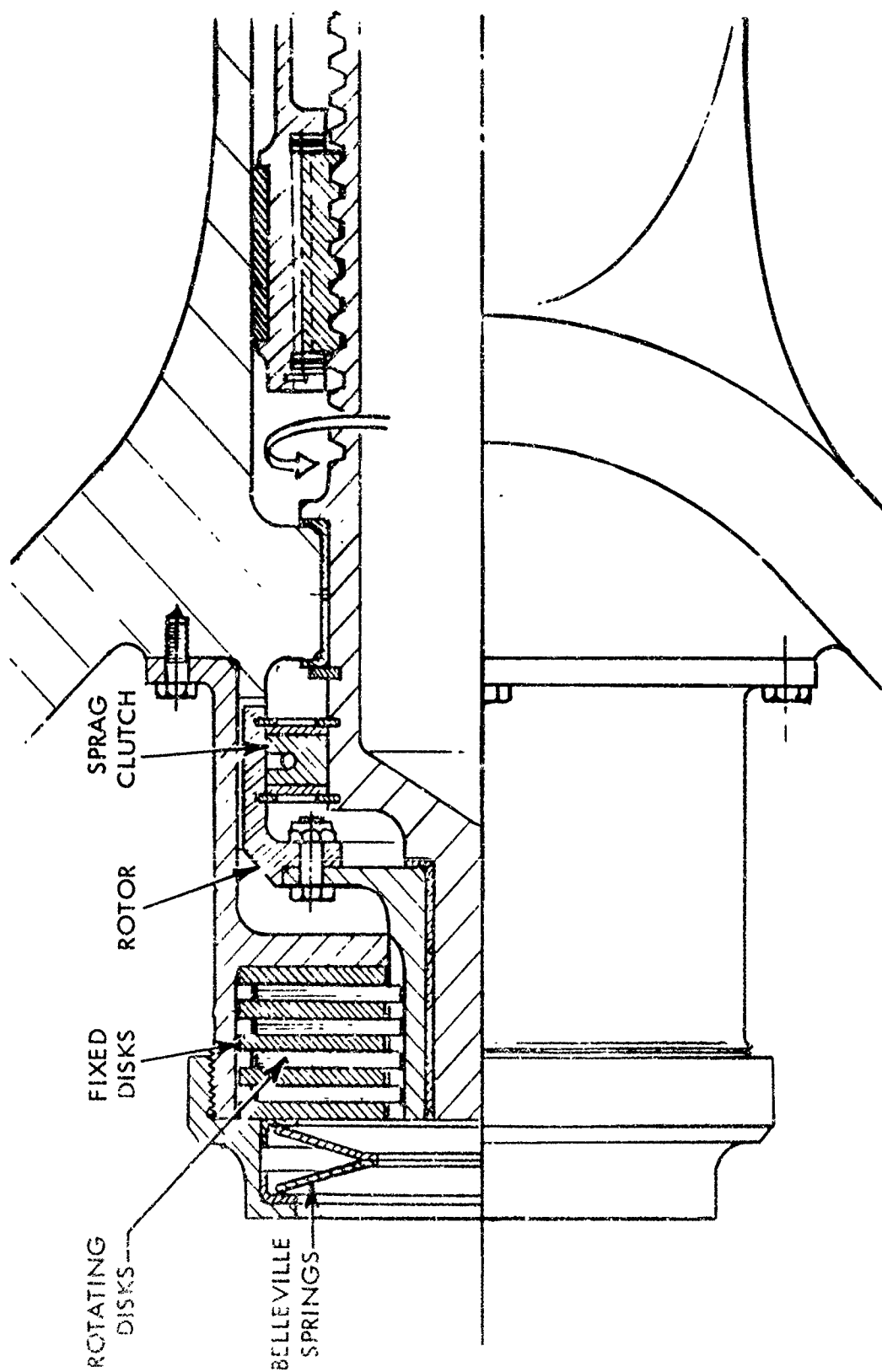


Figure 11. Rotary Friction Shock Absorber

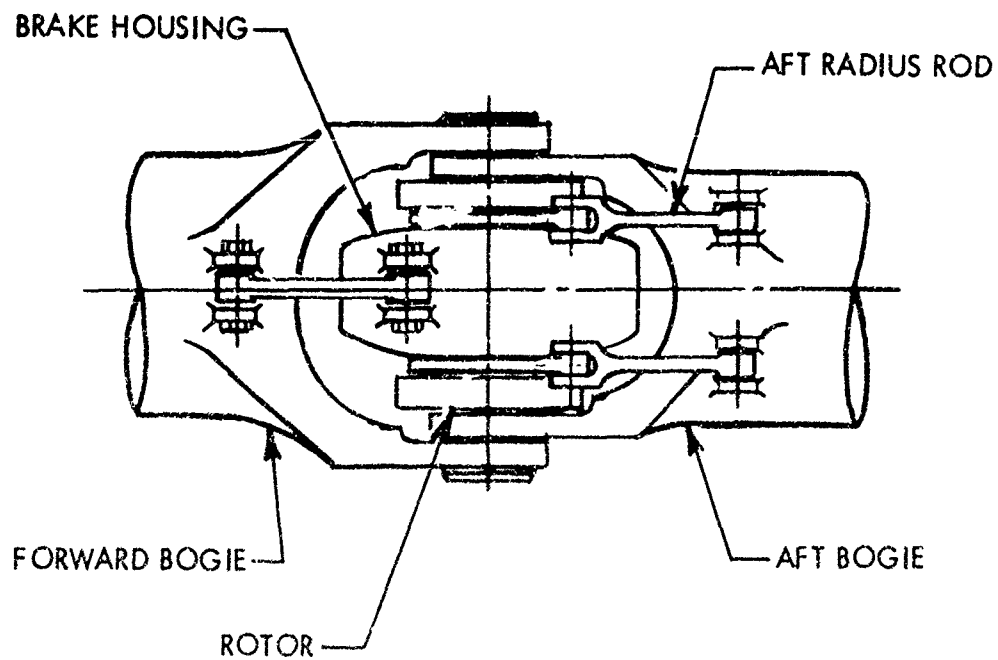
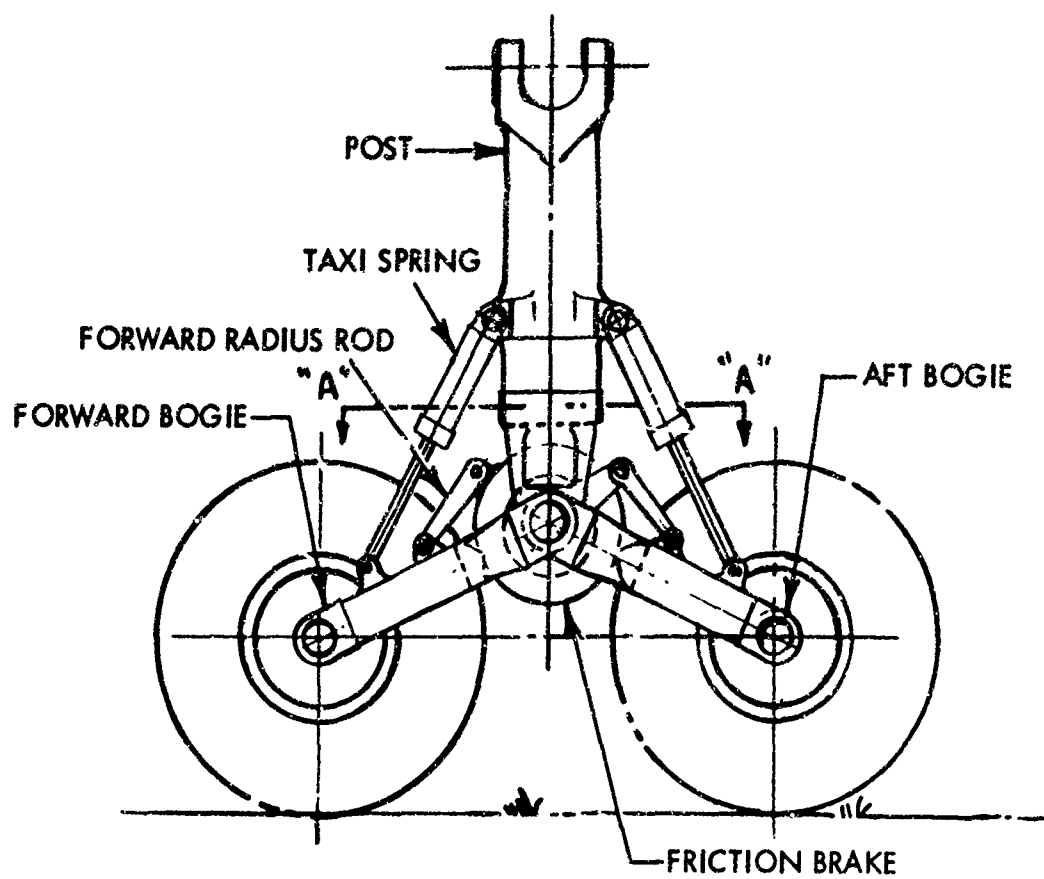


Figure 12. Bogie Type Gear
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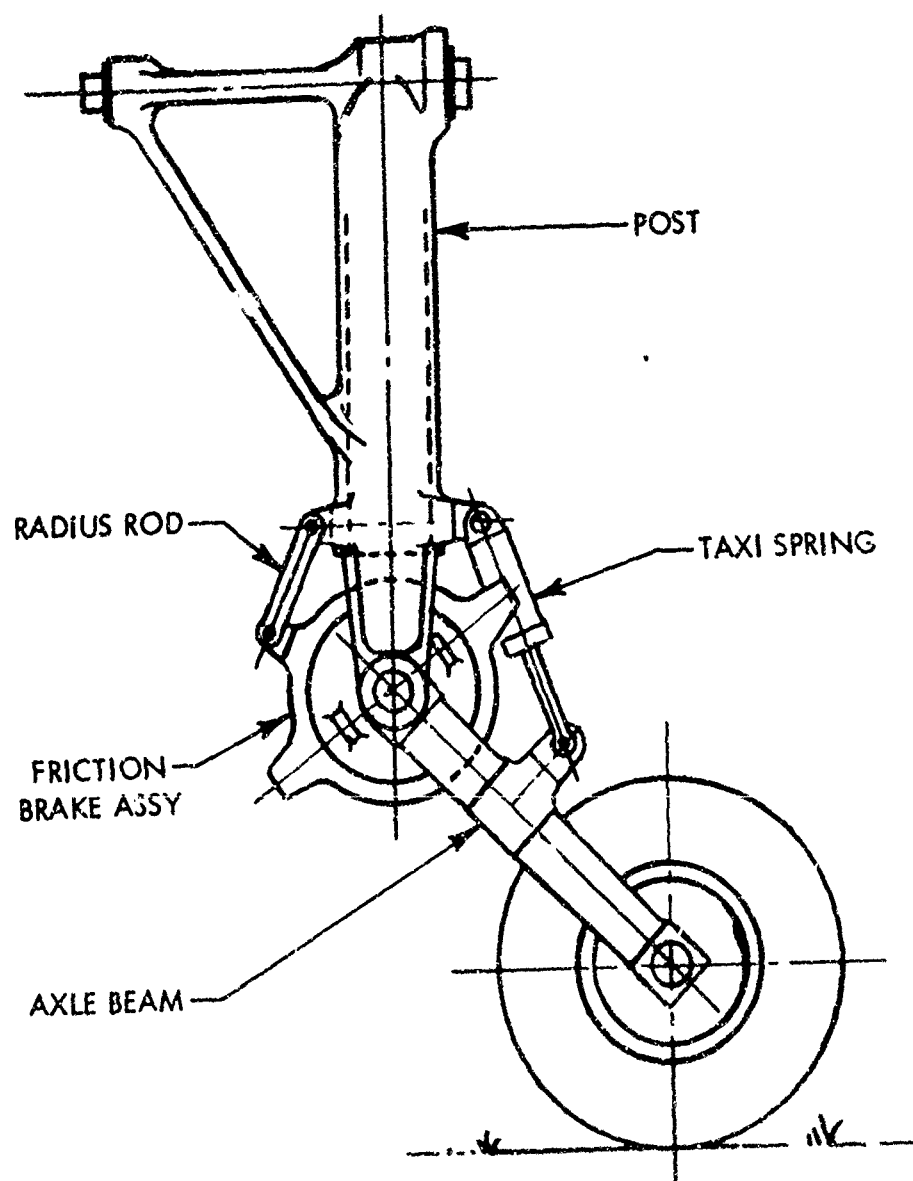


Figure 13. Rotary Friction Brake

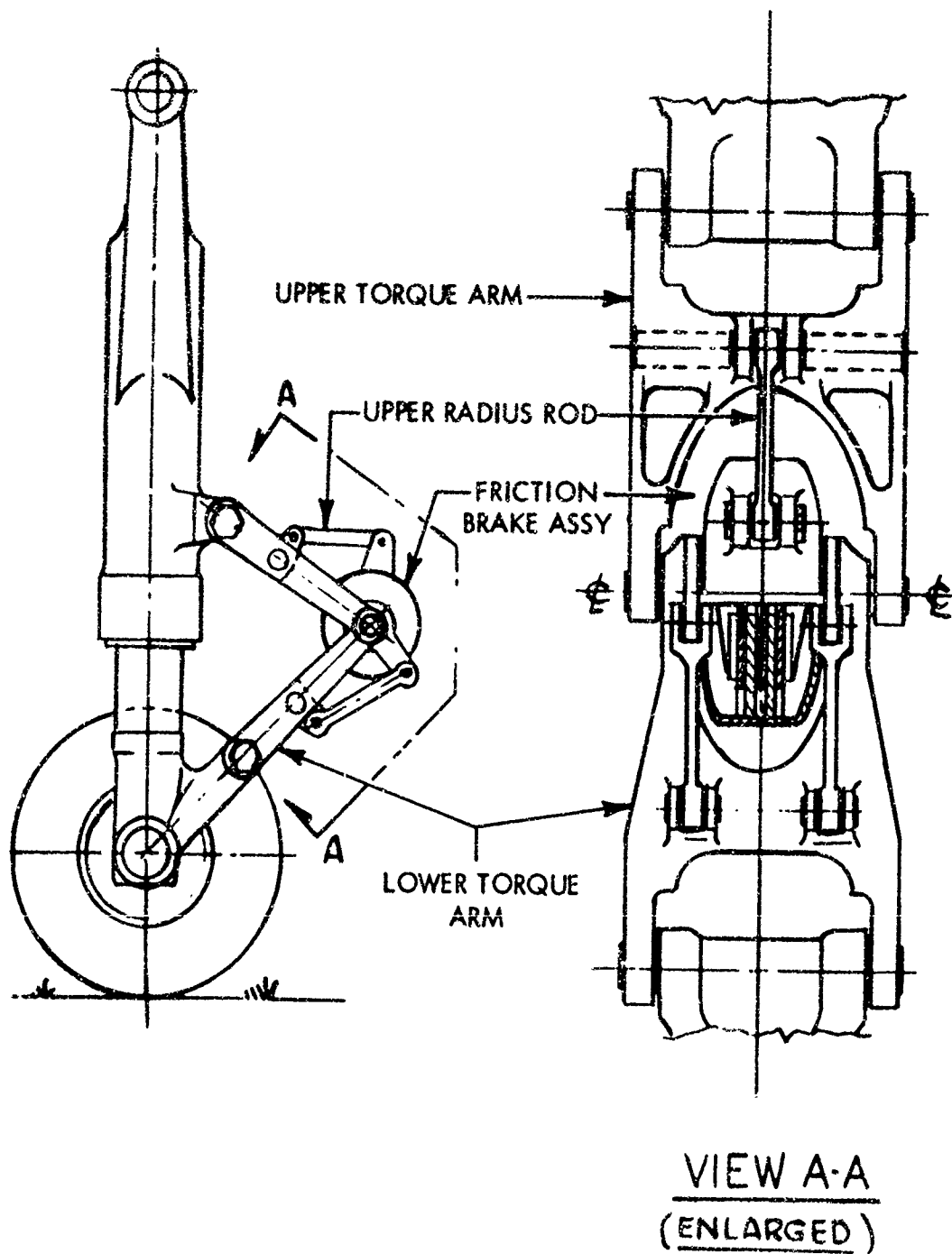


Figure 14. Telescoping Strut

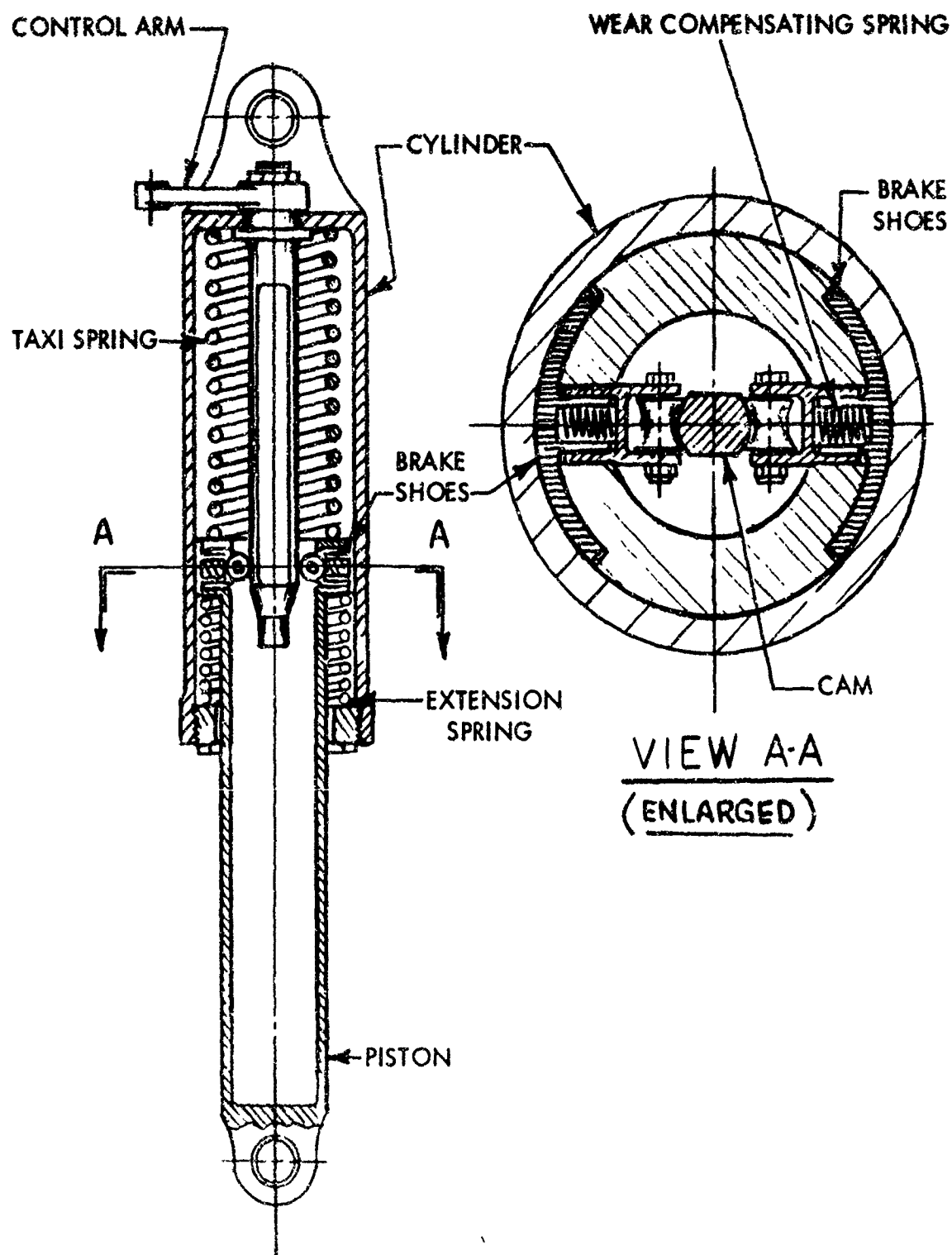


Figure 15. Telescoping Gear "On-Off" Brake Control

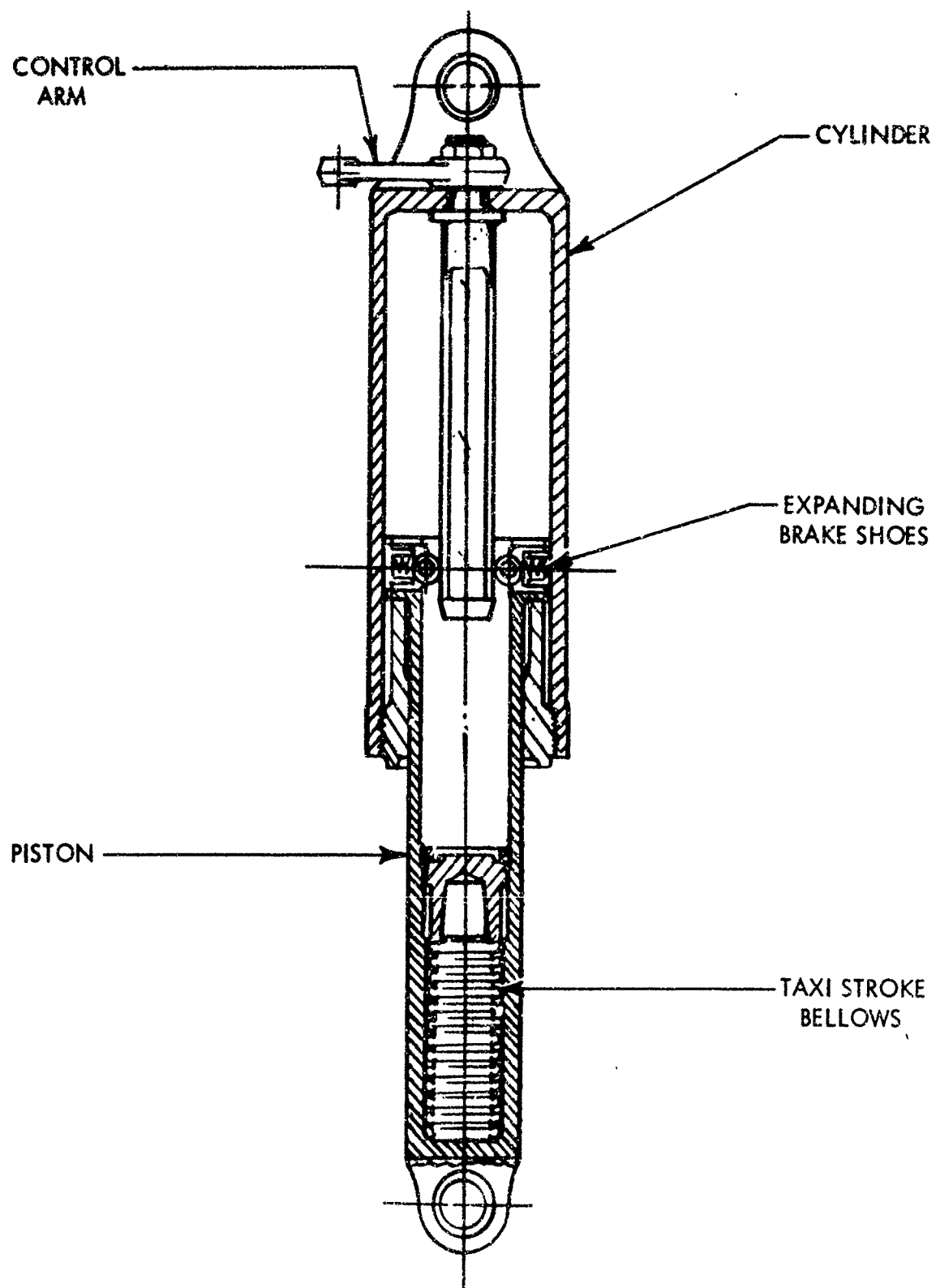


Figure 16. Telescoping Gear - With Bellows

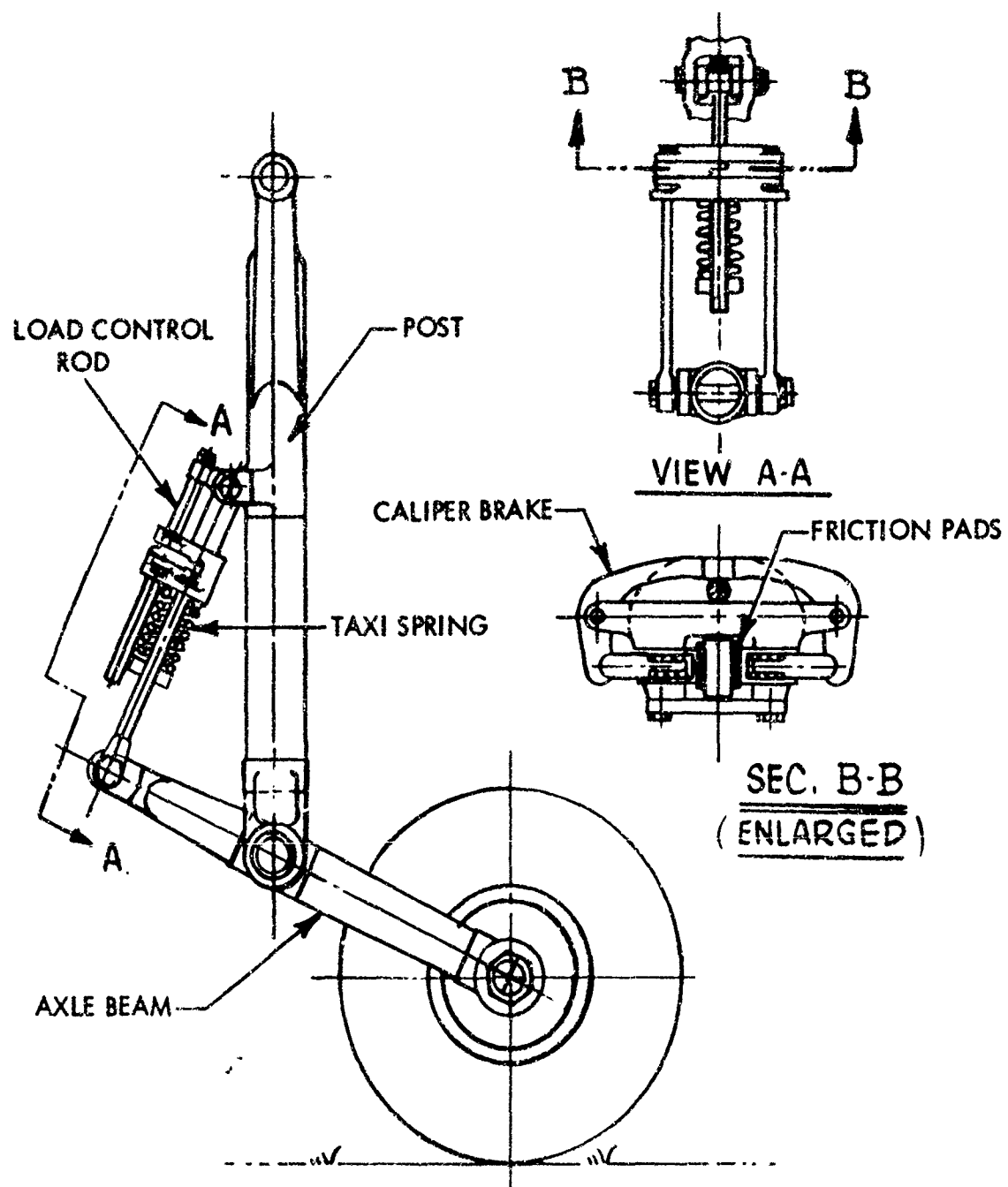


Figure 17. Articulated Strut-Tension Absorber

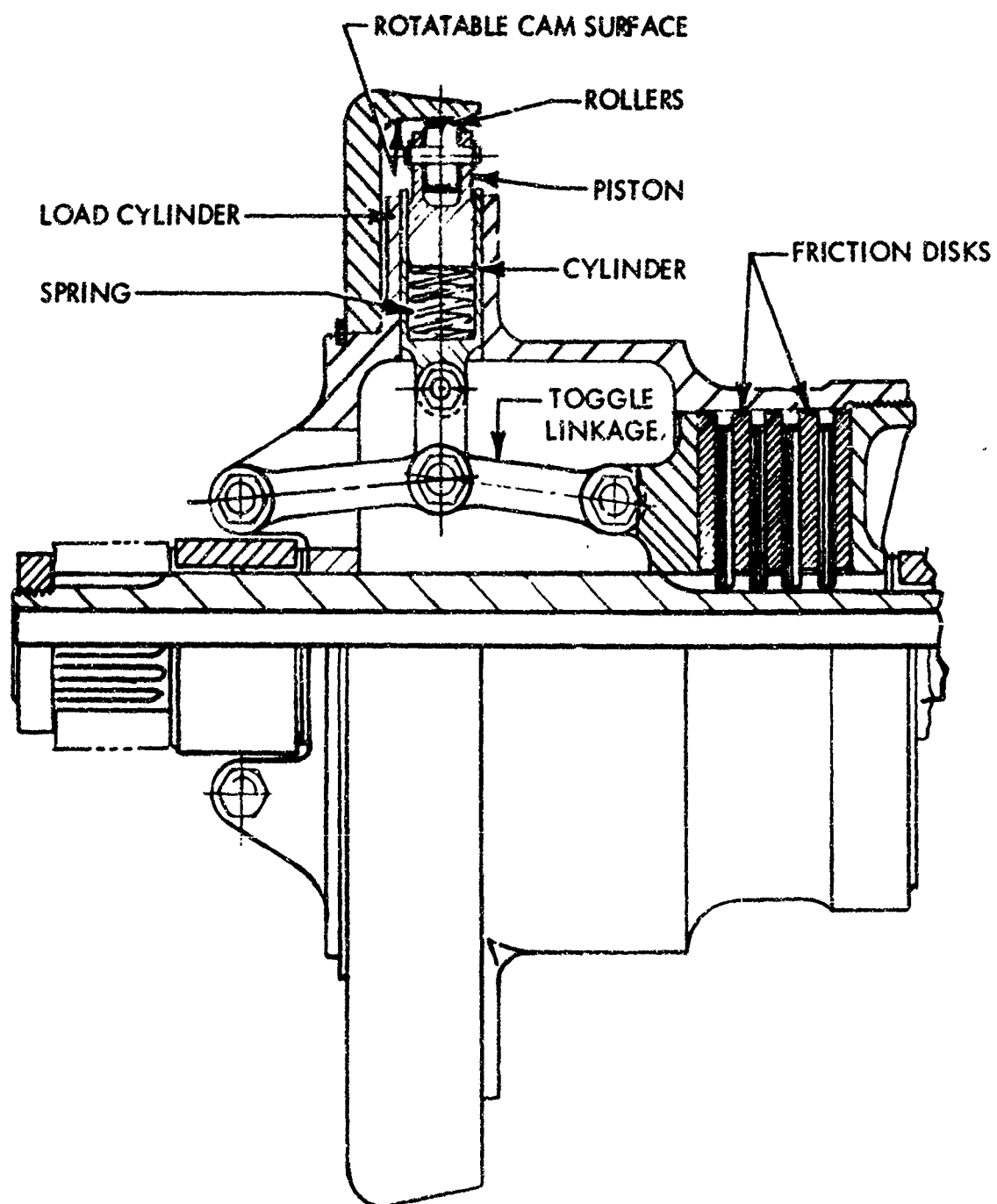


Figure 18. Wear Compensator

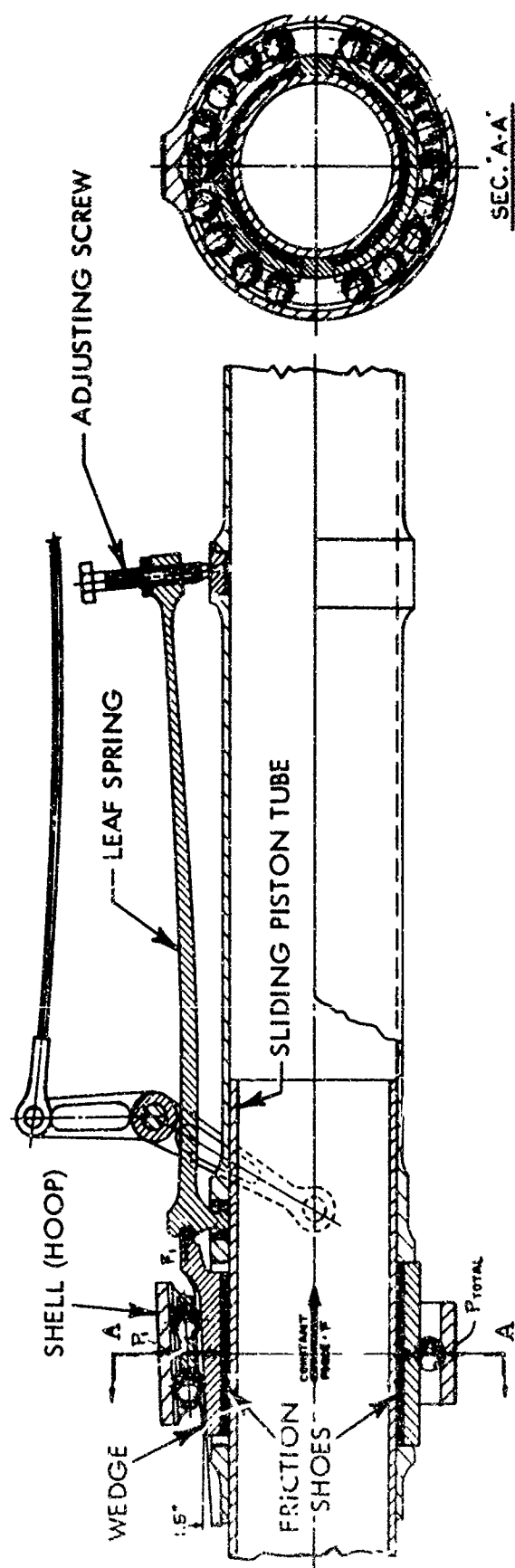


Figure 19. Constant Friction Device - Vtol Aircraft

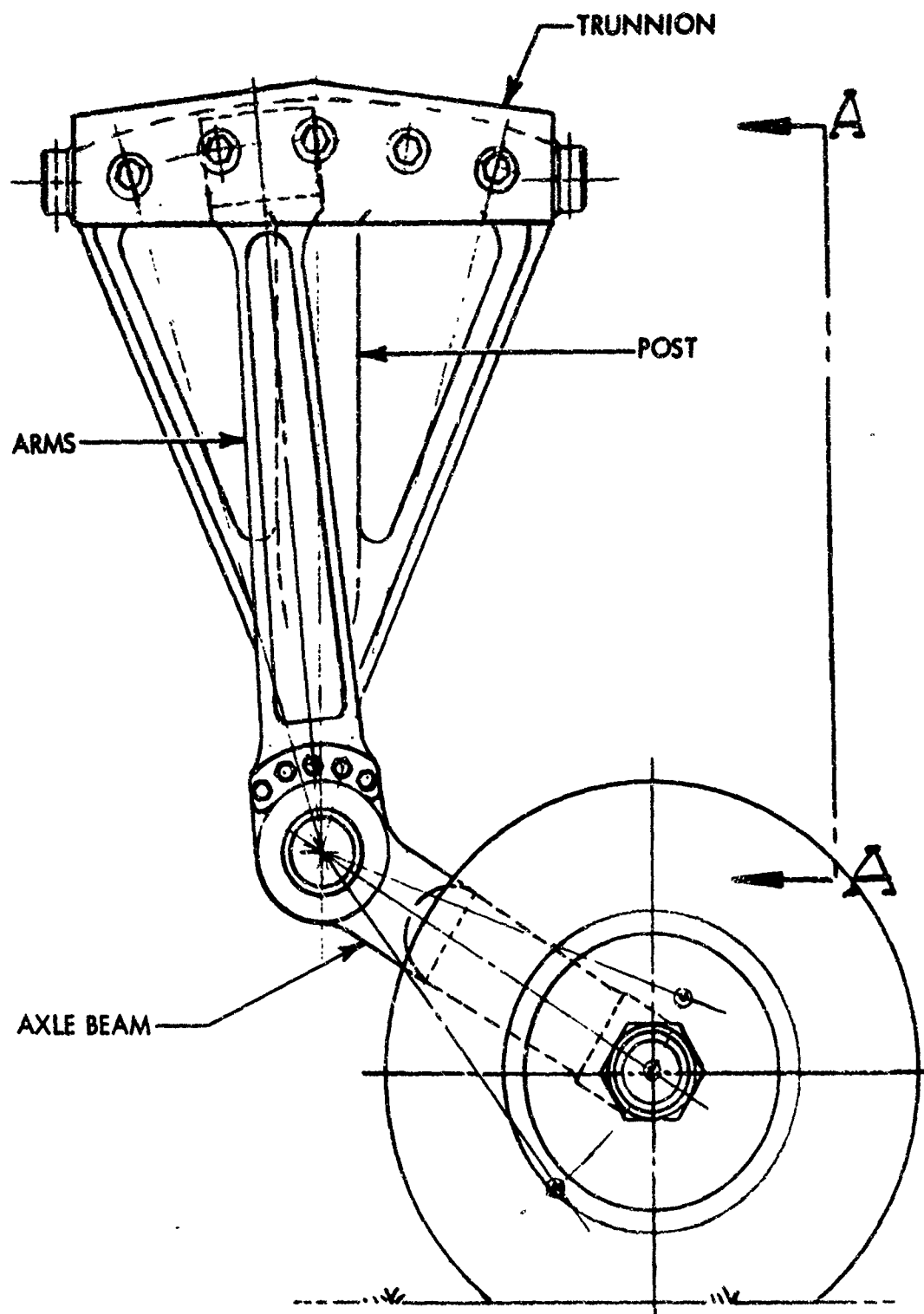


Figure 20. Articulated Gear (Sheet 1 of 2)

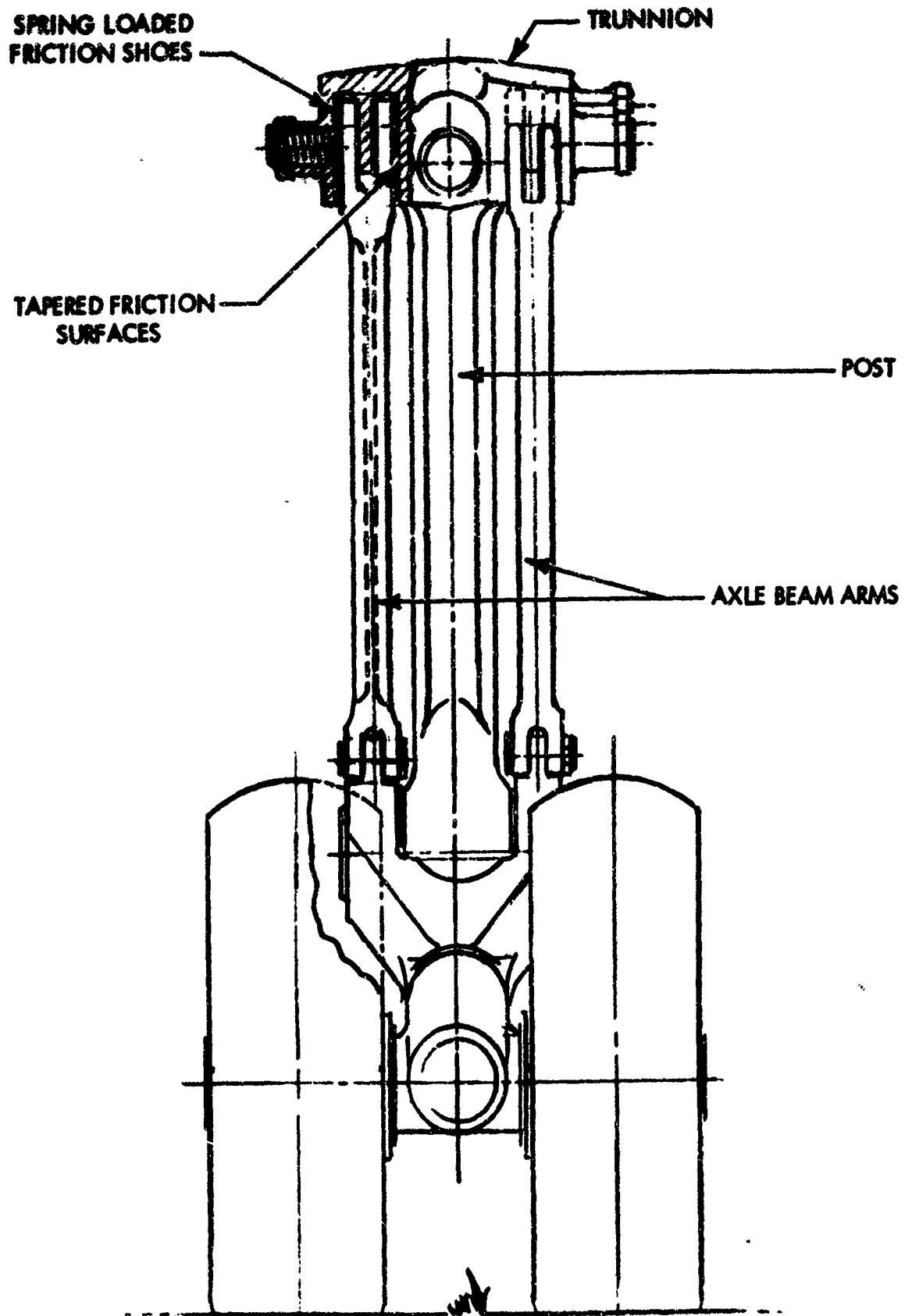


Figure 20. Articulated Gear (Sheet 2 of 2)
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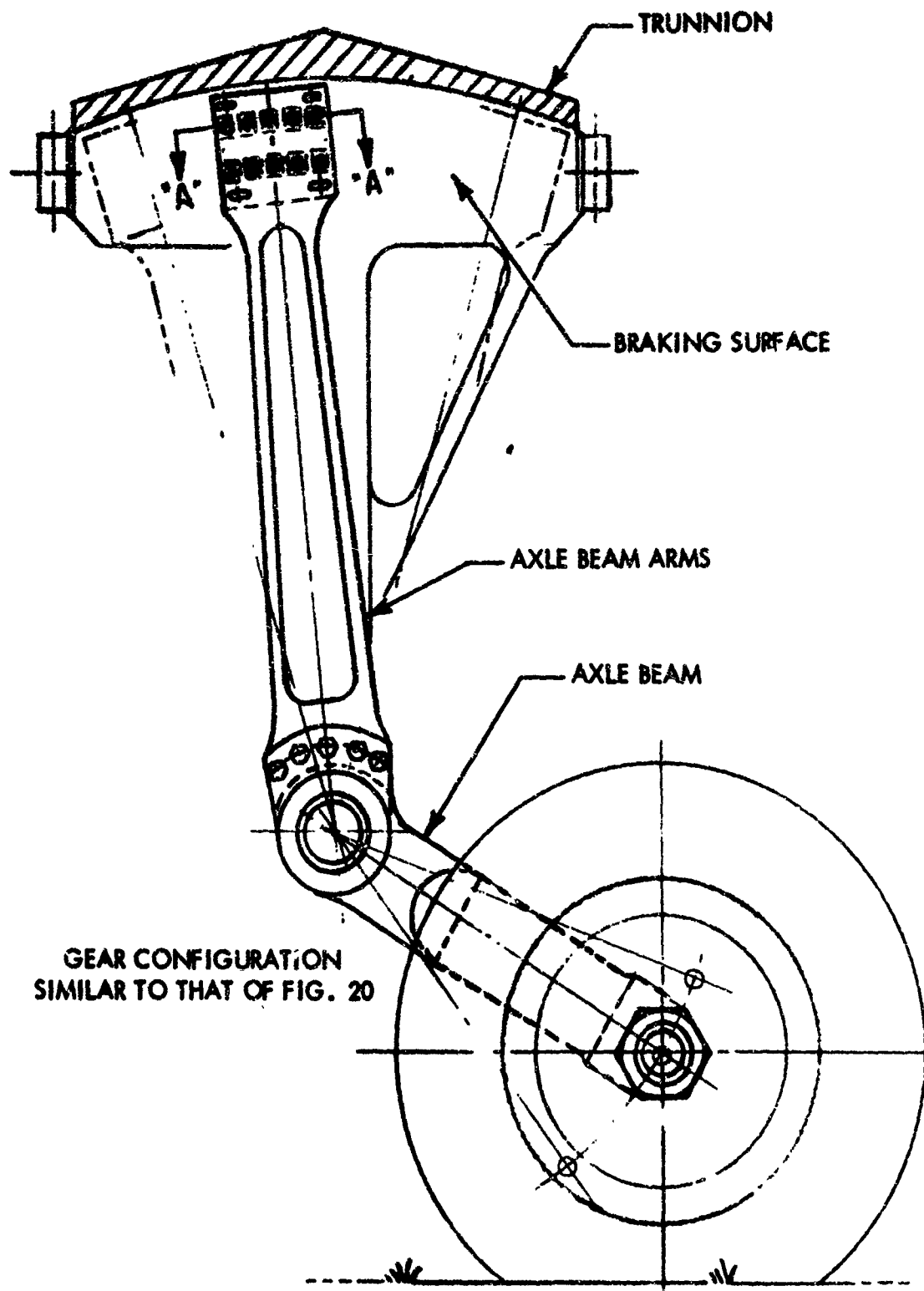


Figure 21. Automatic One-Way Brake (Sheet 1 of 2)

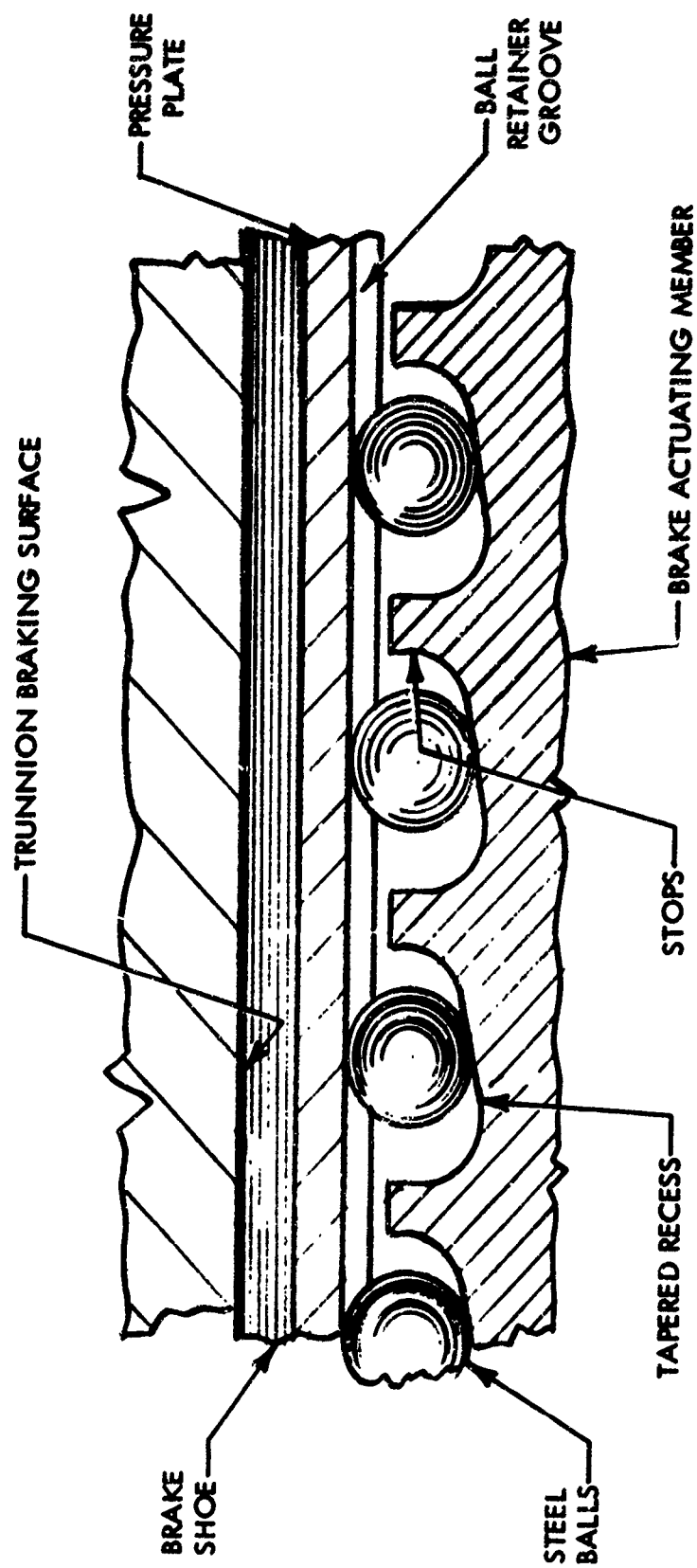


Figure 21. Automatic One-Way Brake (Sheet 2 of 2)
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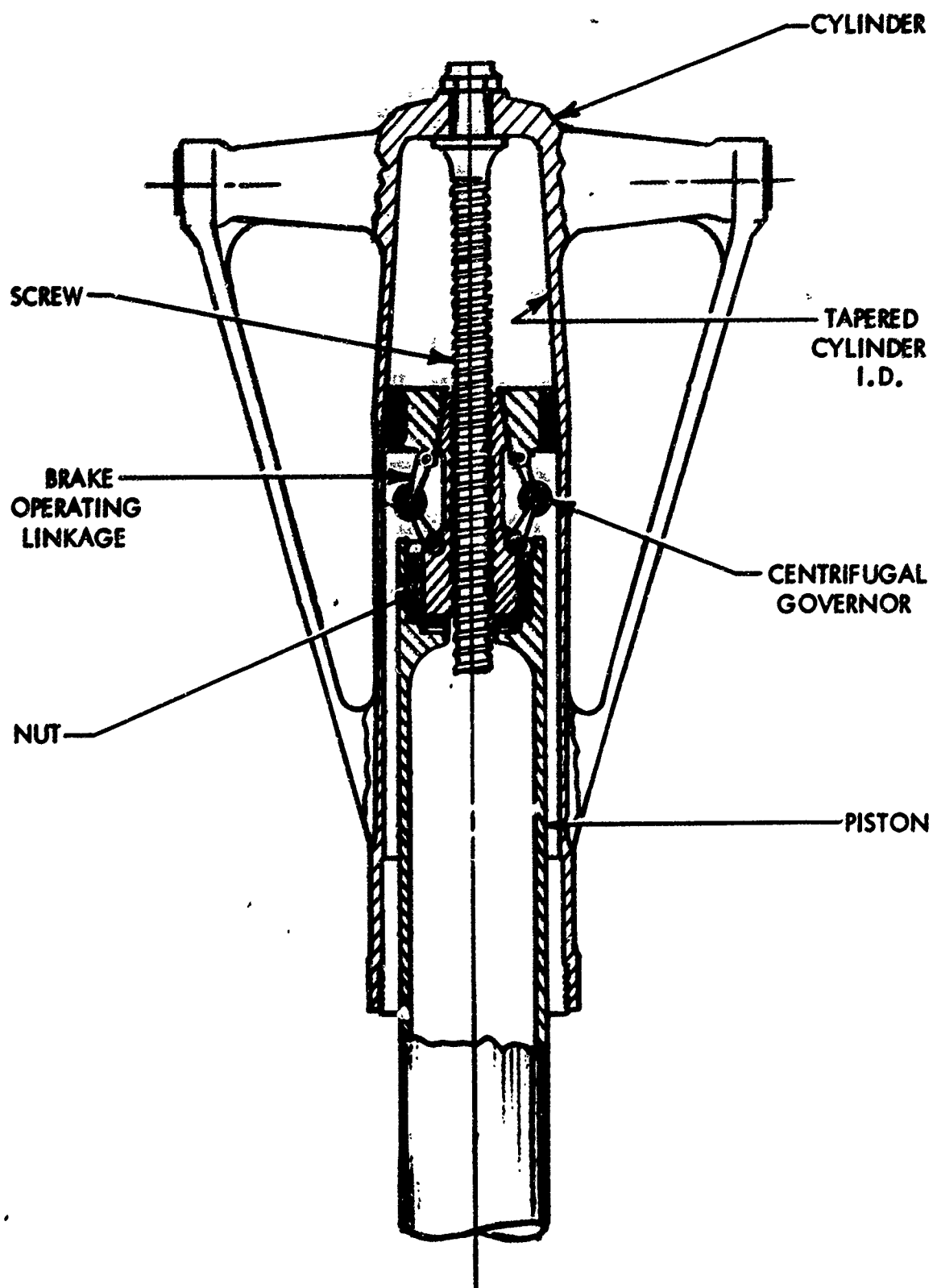


Figure 22. Constant Friction Telescoping Gear

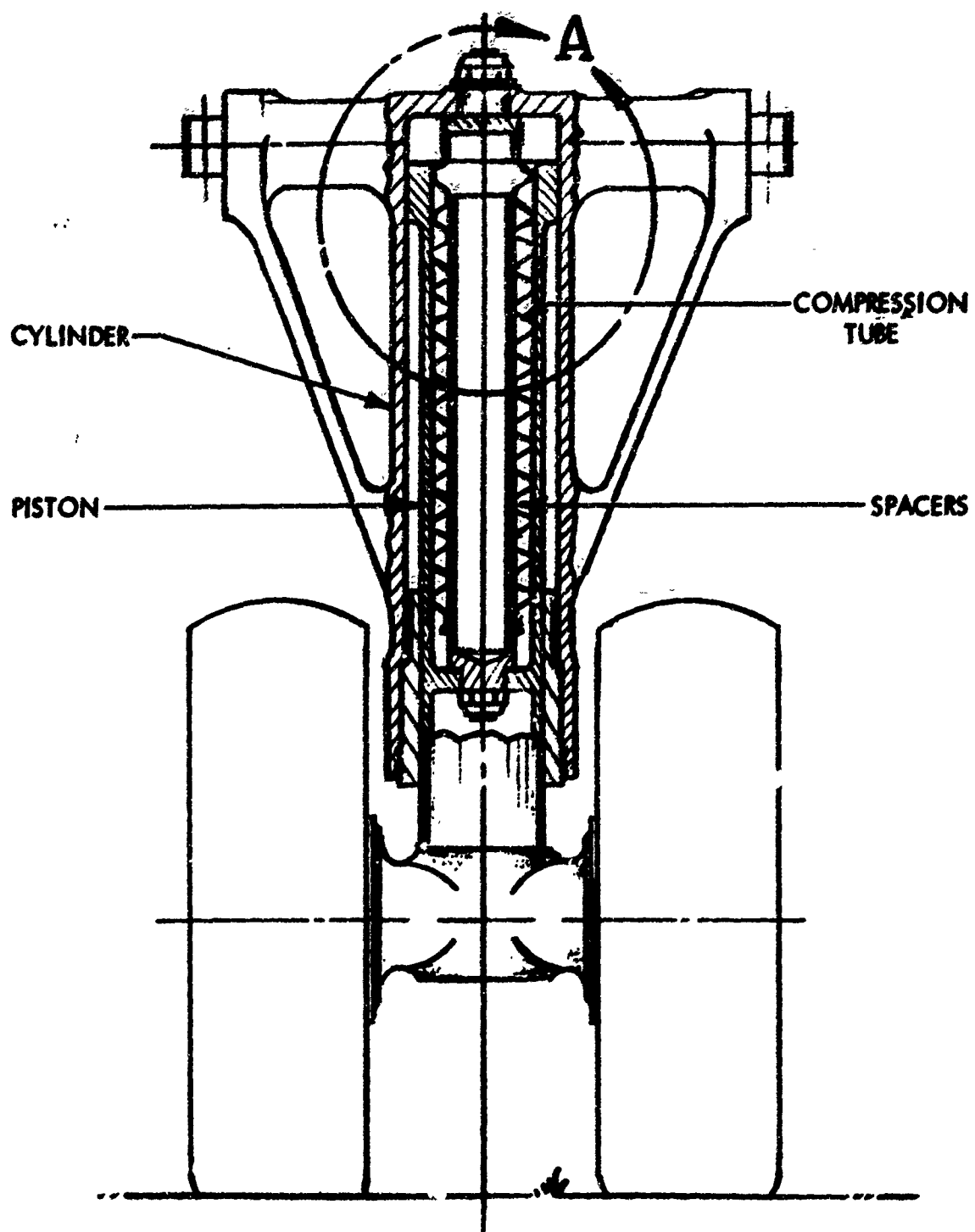


Figure 23. Constant Friction Gear (Sheet 1 of 2)

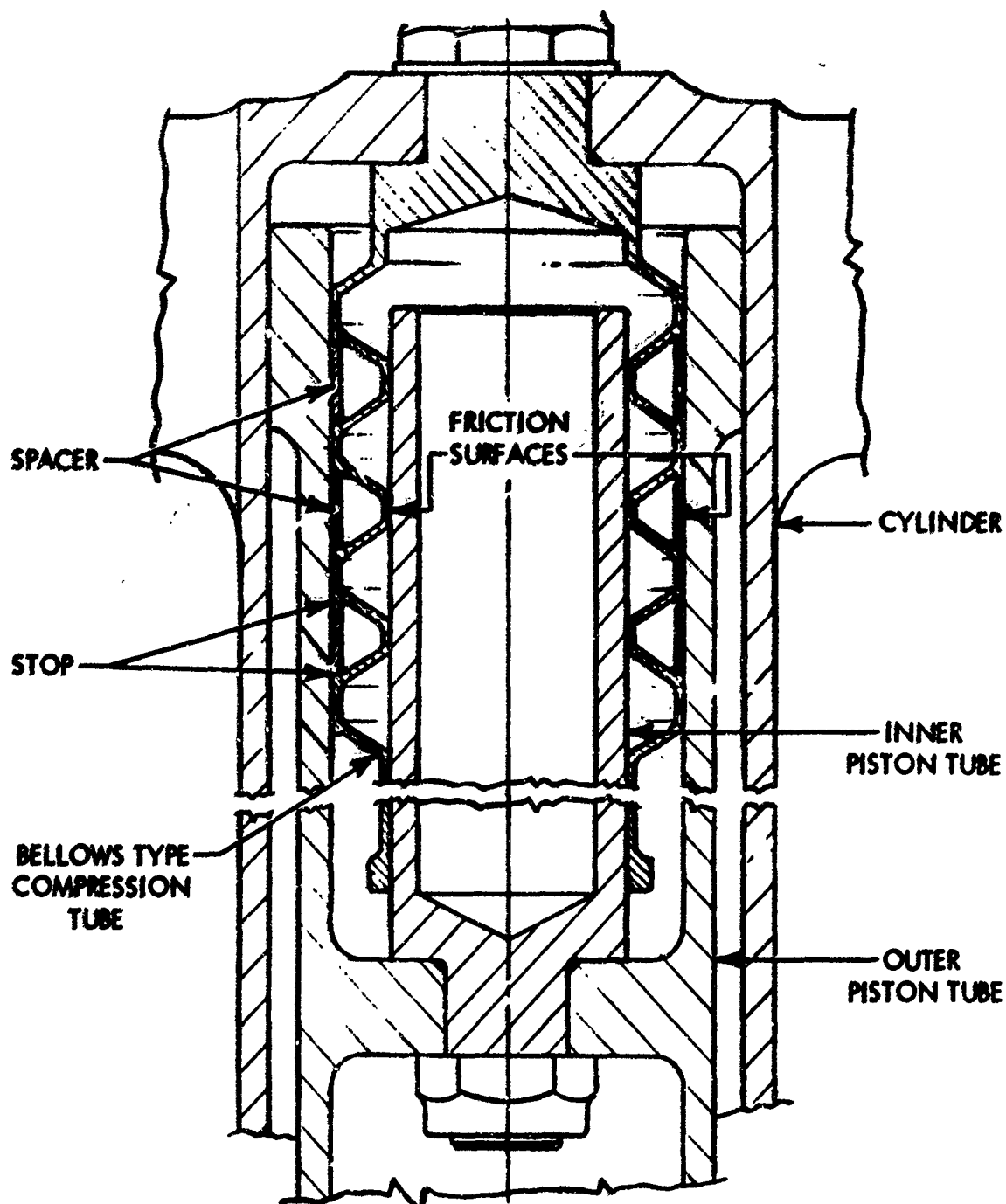


Figure 23. Constant Friction Gear (Sheet 2 of 2)
VIEW "A" (ENLARGED)

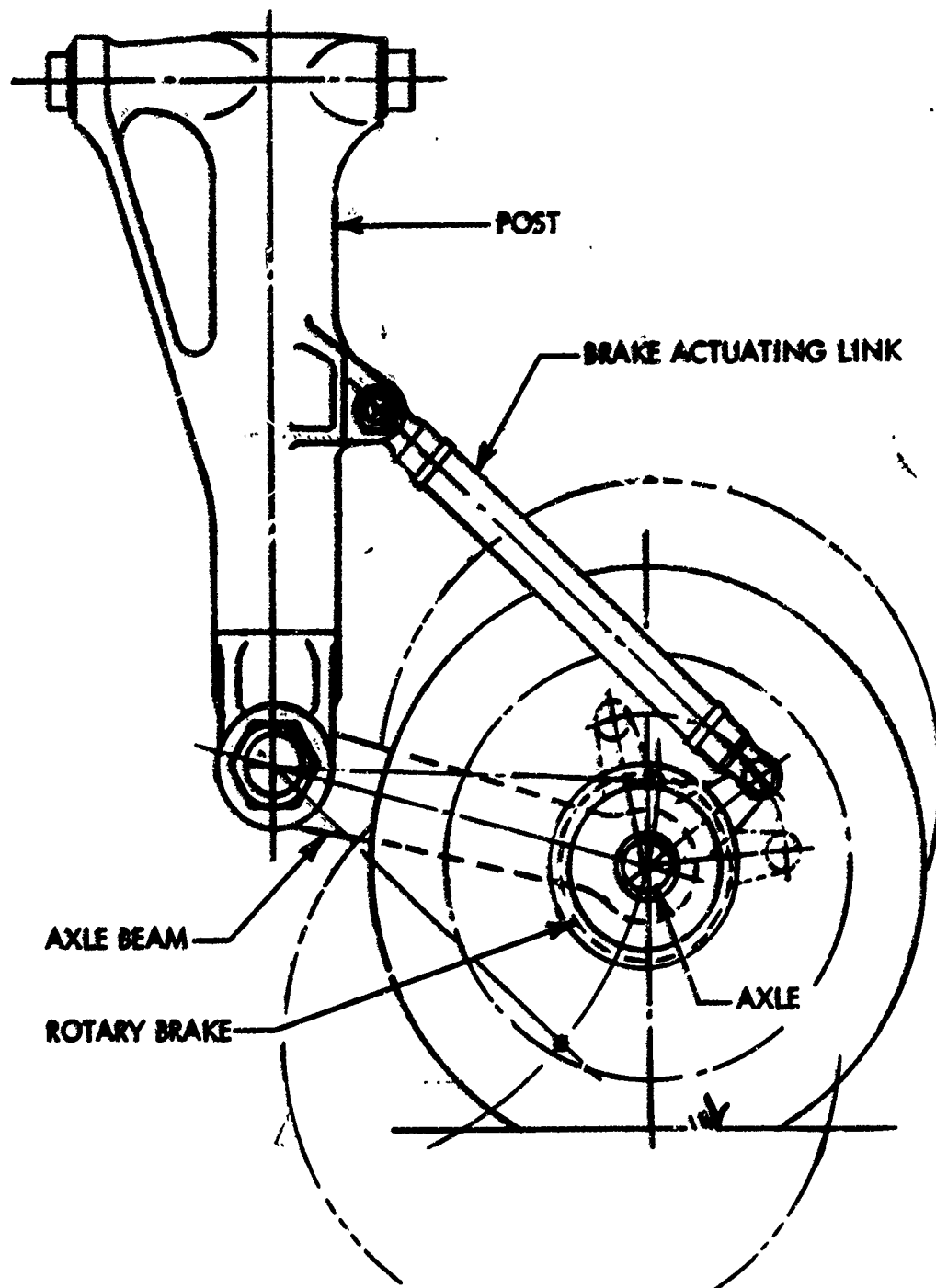


Figure 24. Articulated Gear - Rotary Shock Absorber

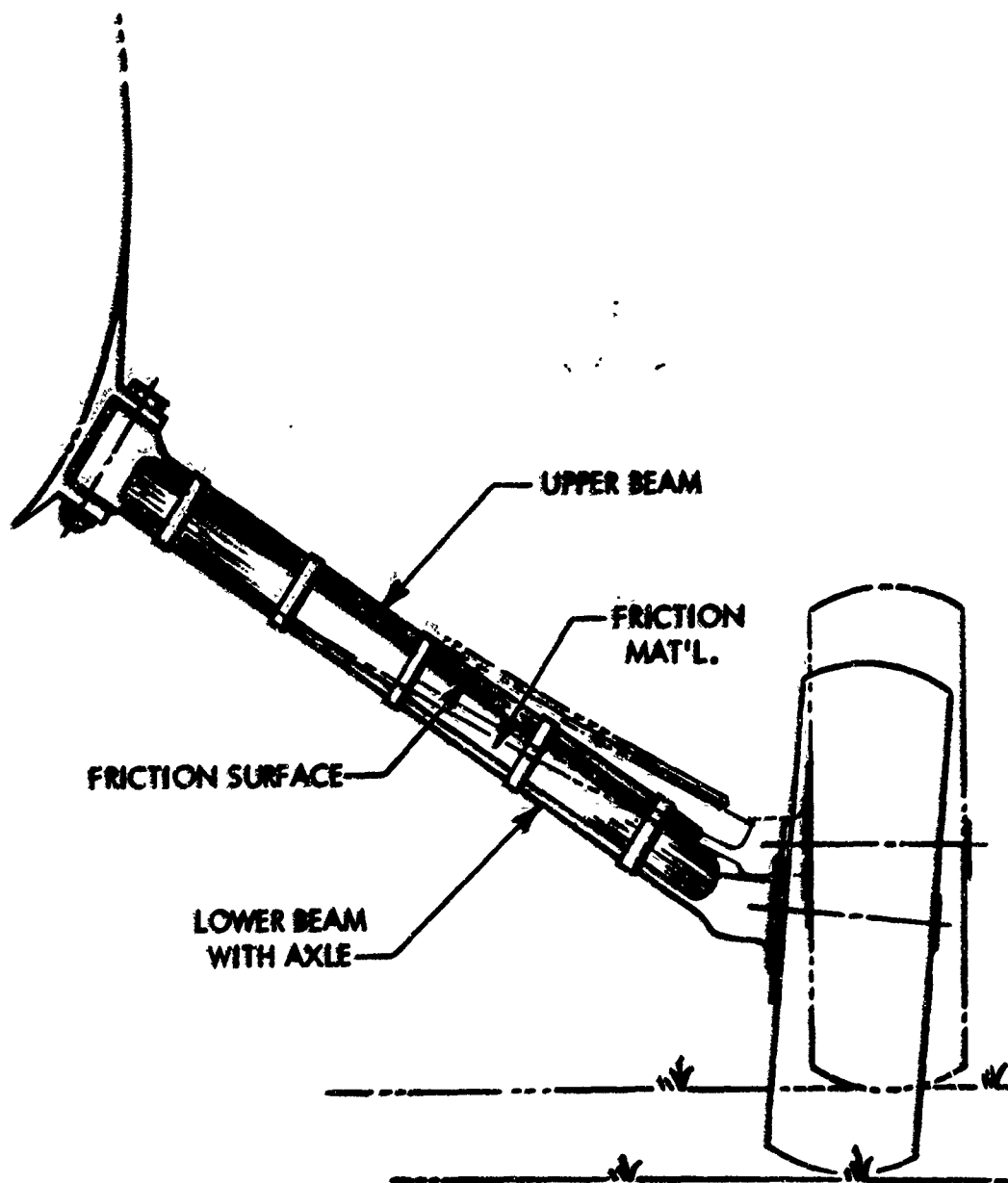


Figure 25. Side Mounted Articulated Gear

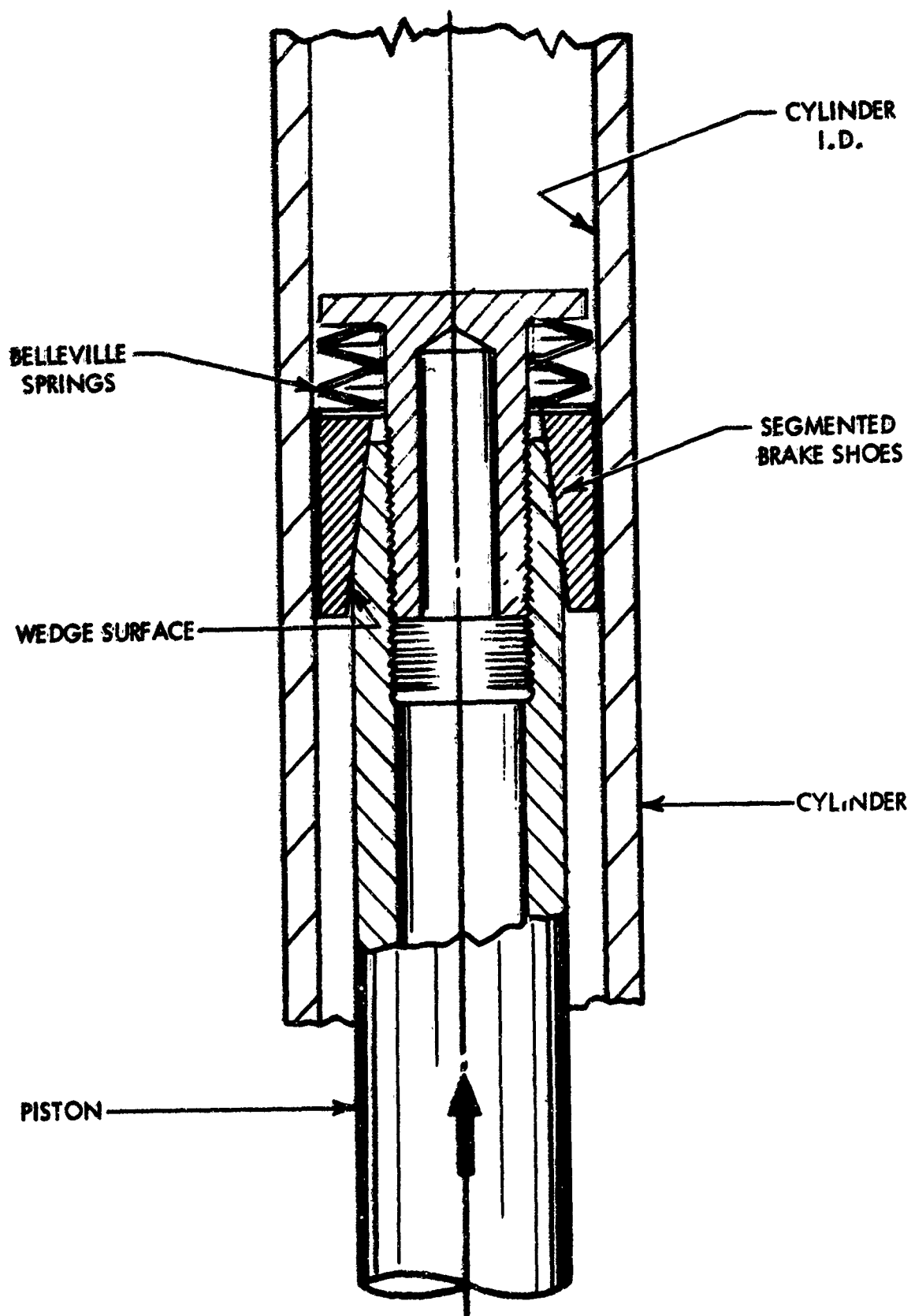


Figure 26. Compression Type Shock Absorber

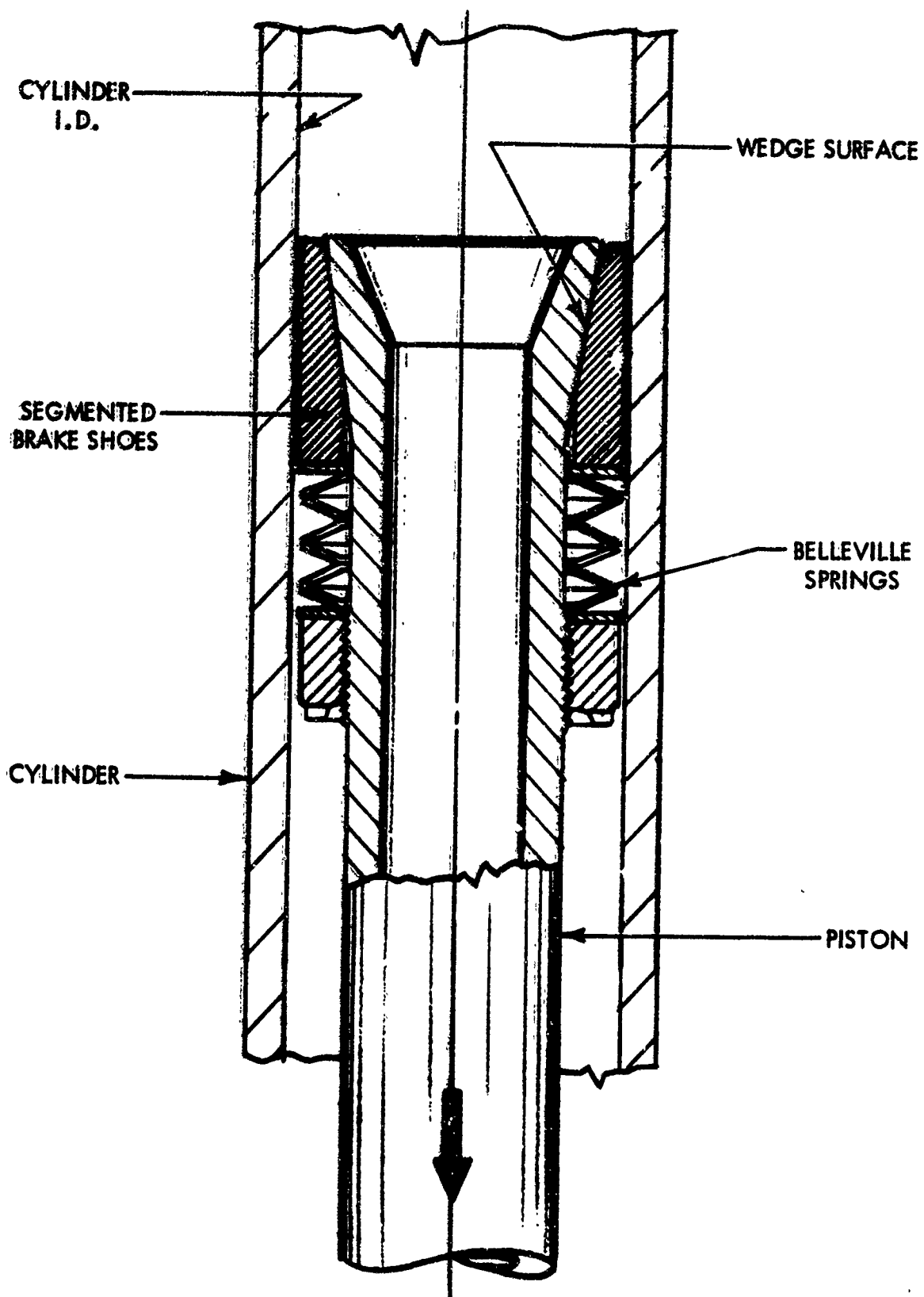


Figure 27. Tension Type Shock Absorber

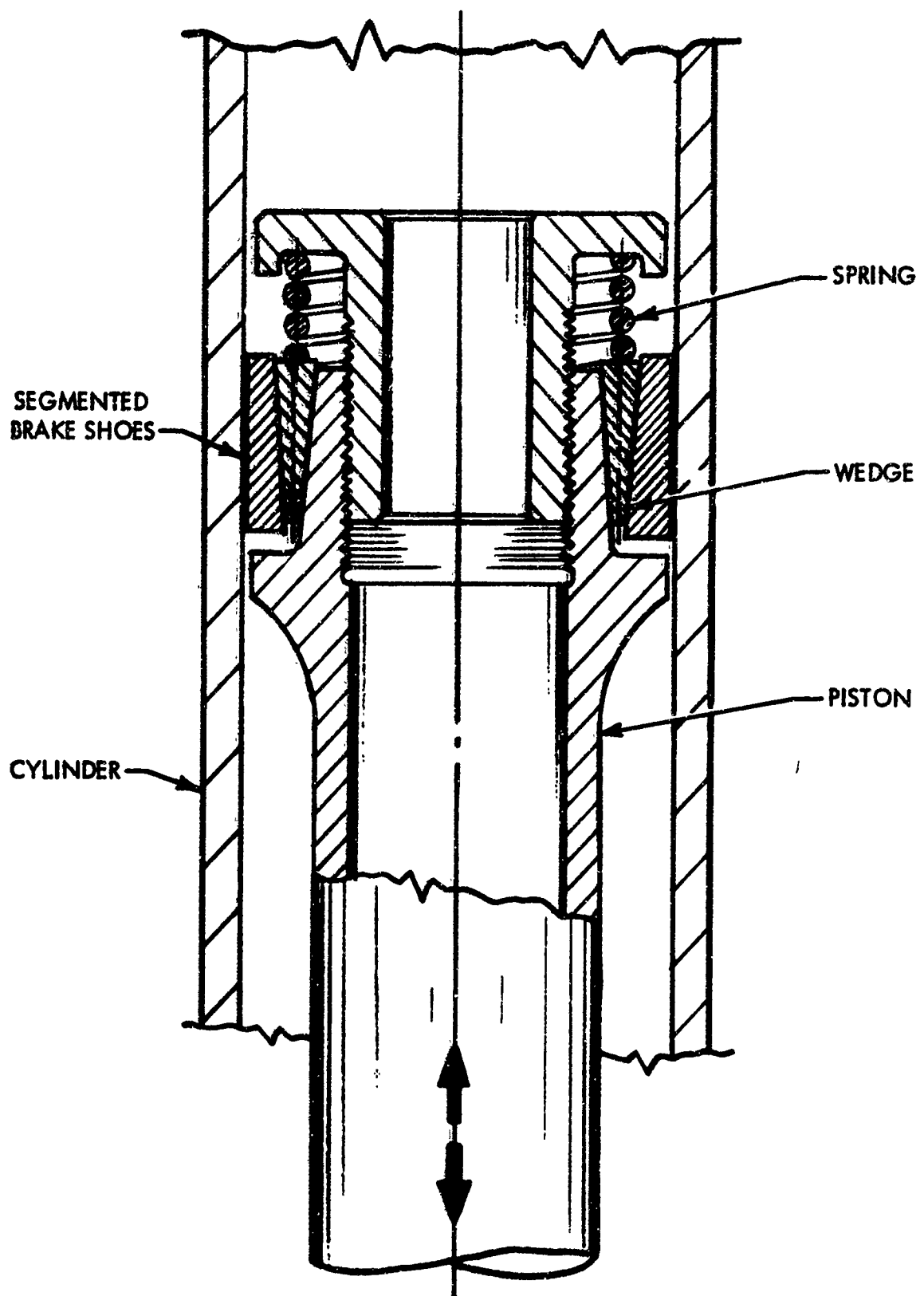


Figure 28. Constant Friction Shock Absorber

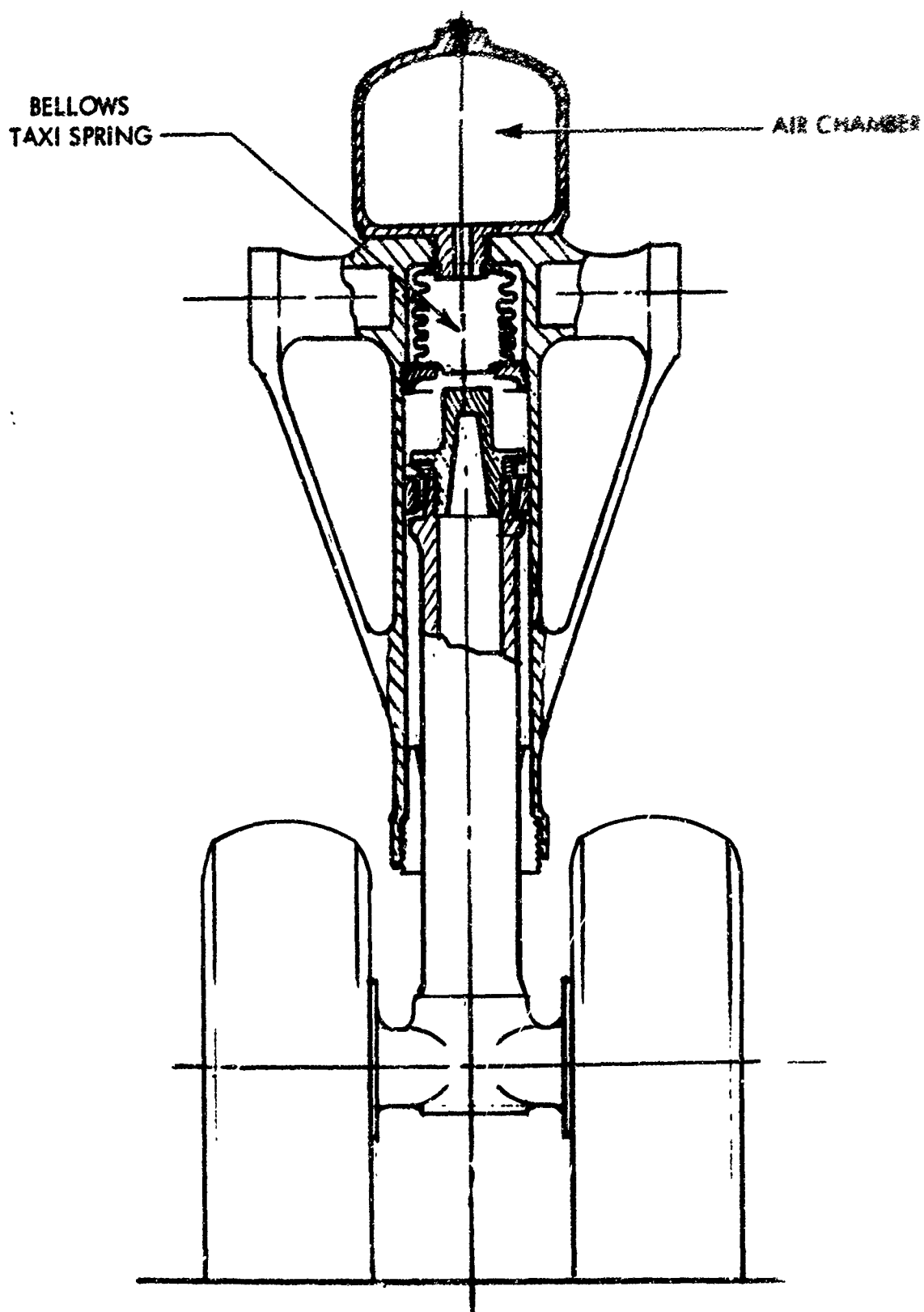


Figure 29. Constant Friction Shock Absorber and Bellows

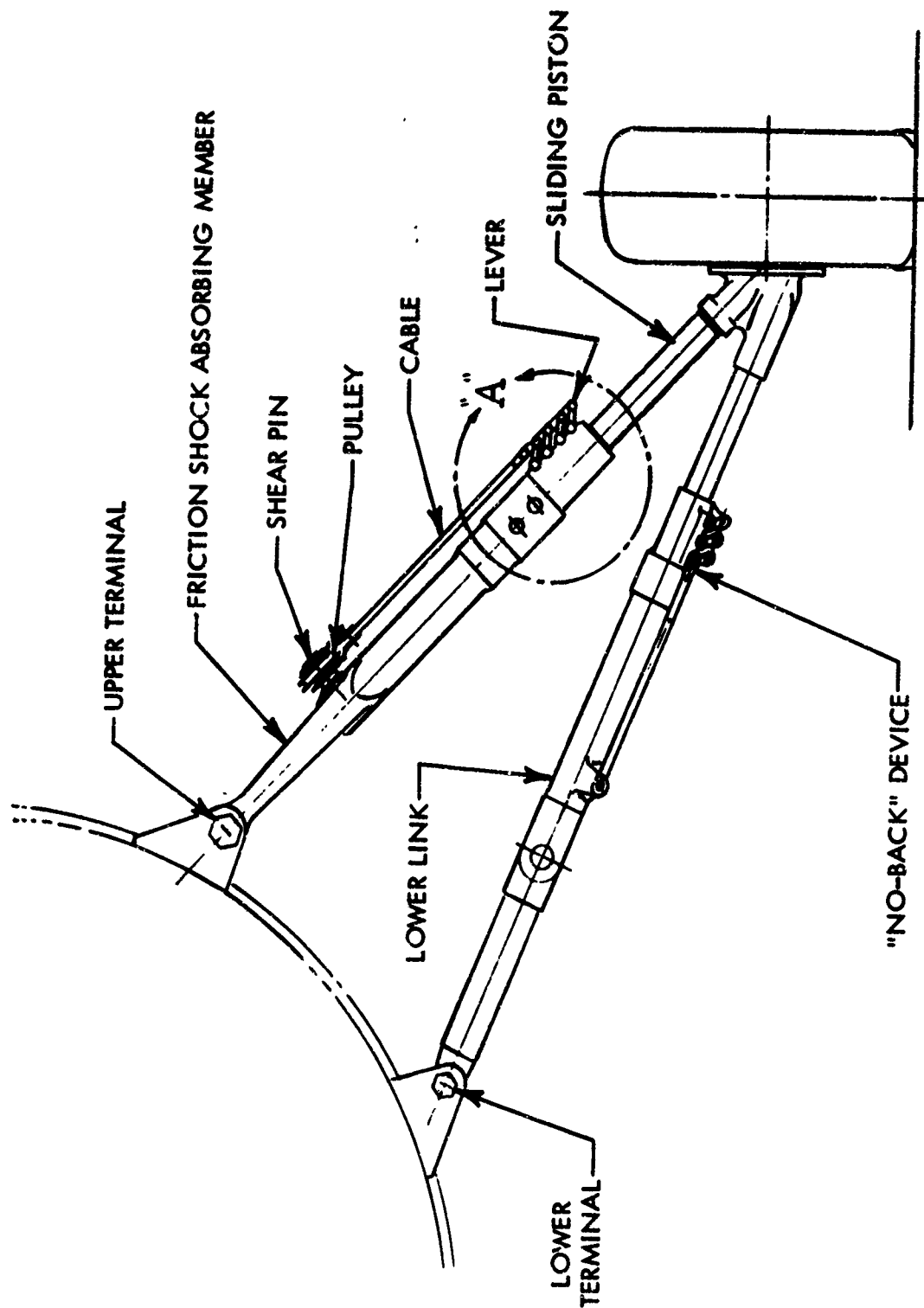


Figure 30. Vtol Type, Tripod Landing Gear (Sheet 1 of 2)

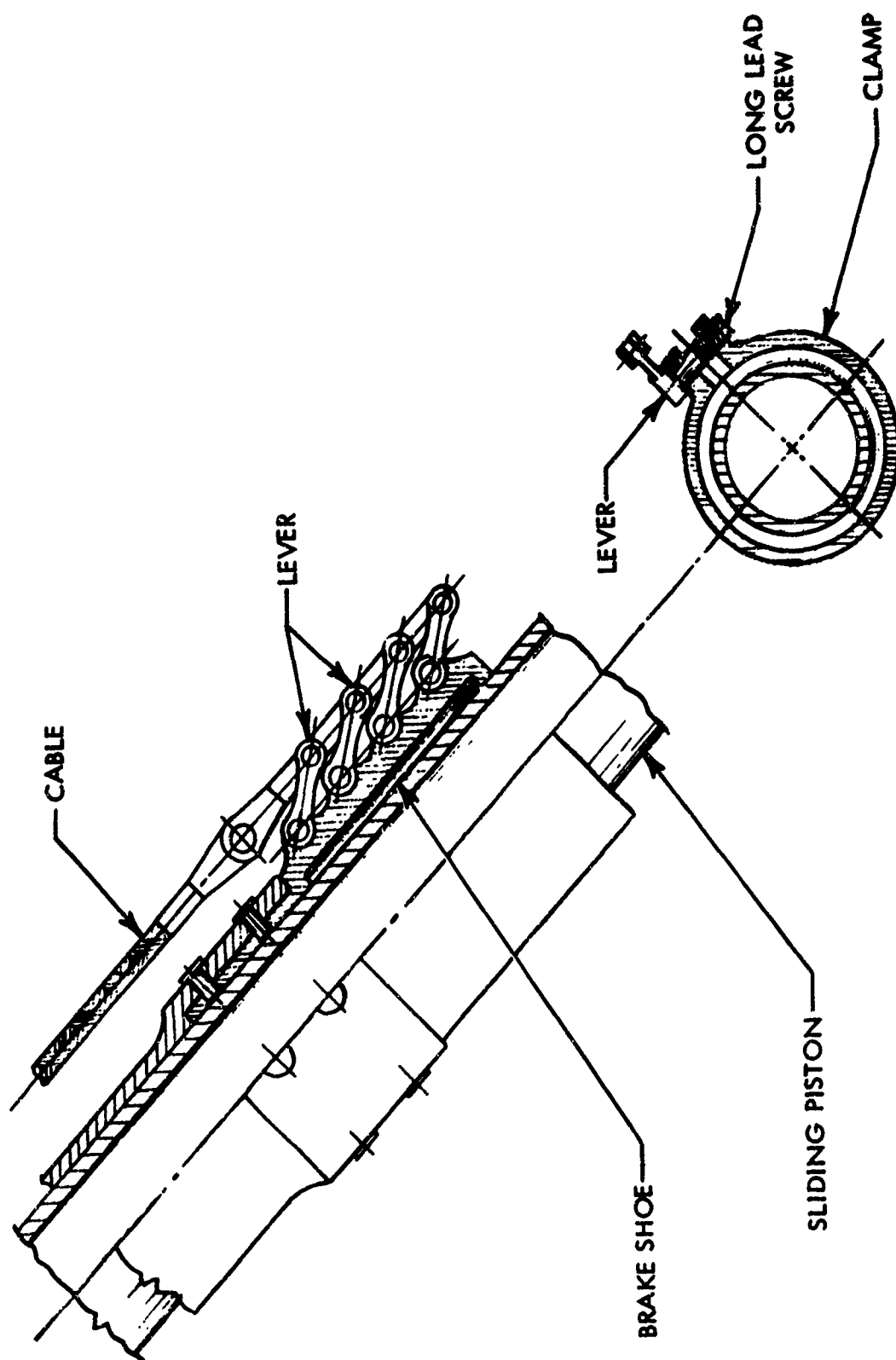


Figure 30. Vtol Type, Tripod Landing Gear (Sheet 2 of 2)
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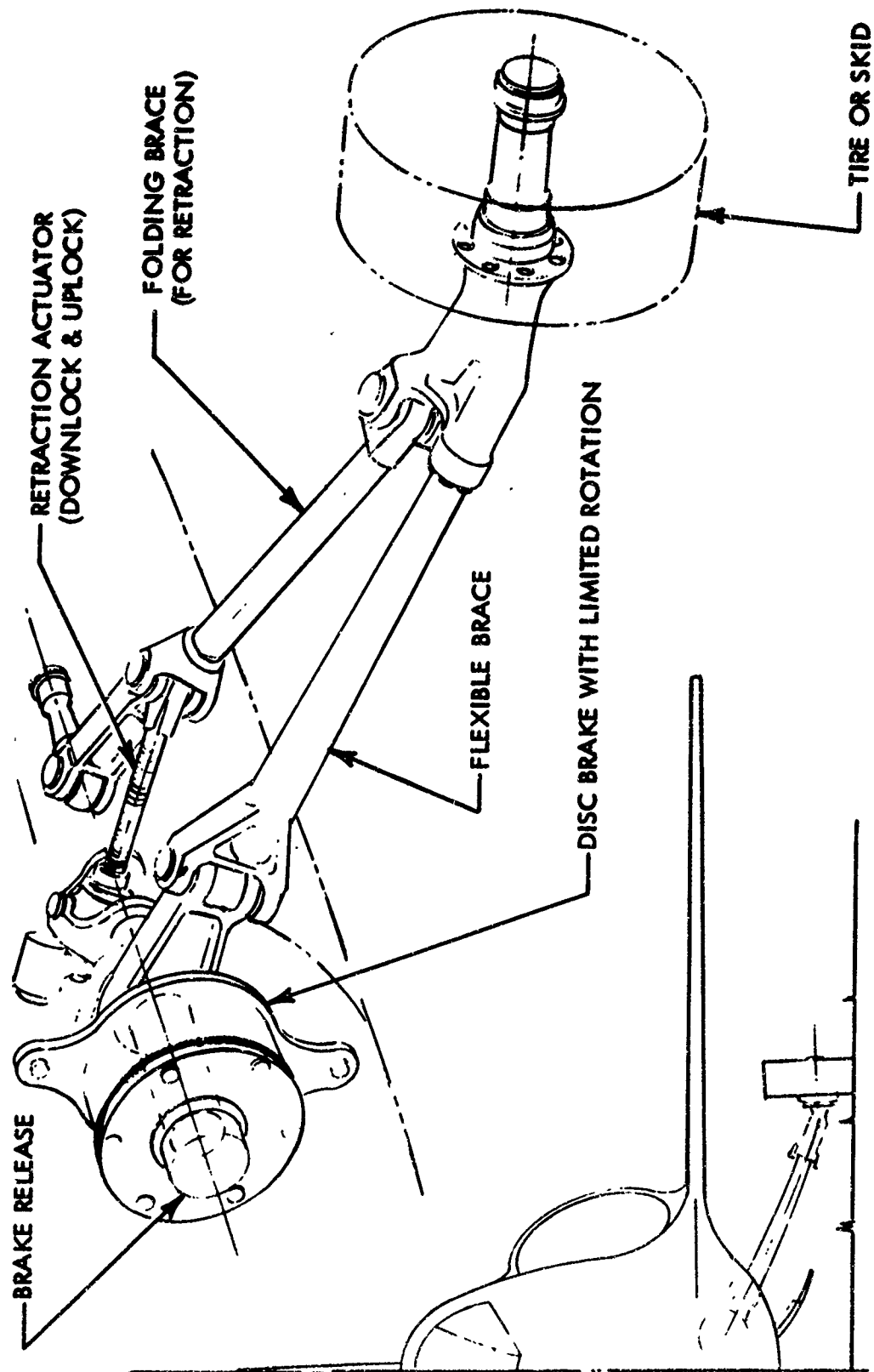


Figure 31. Articulated Gear - Rotary Brake

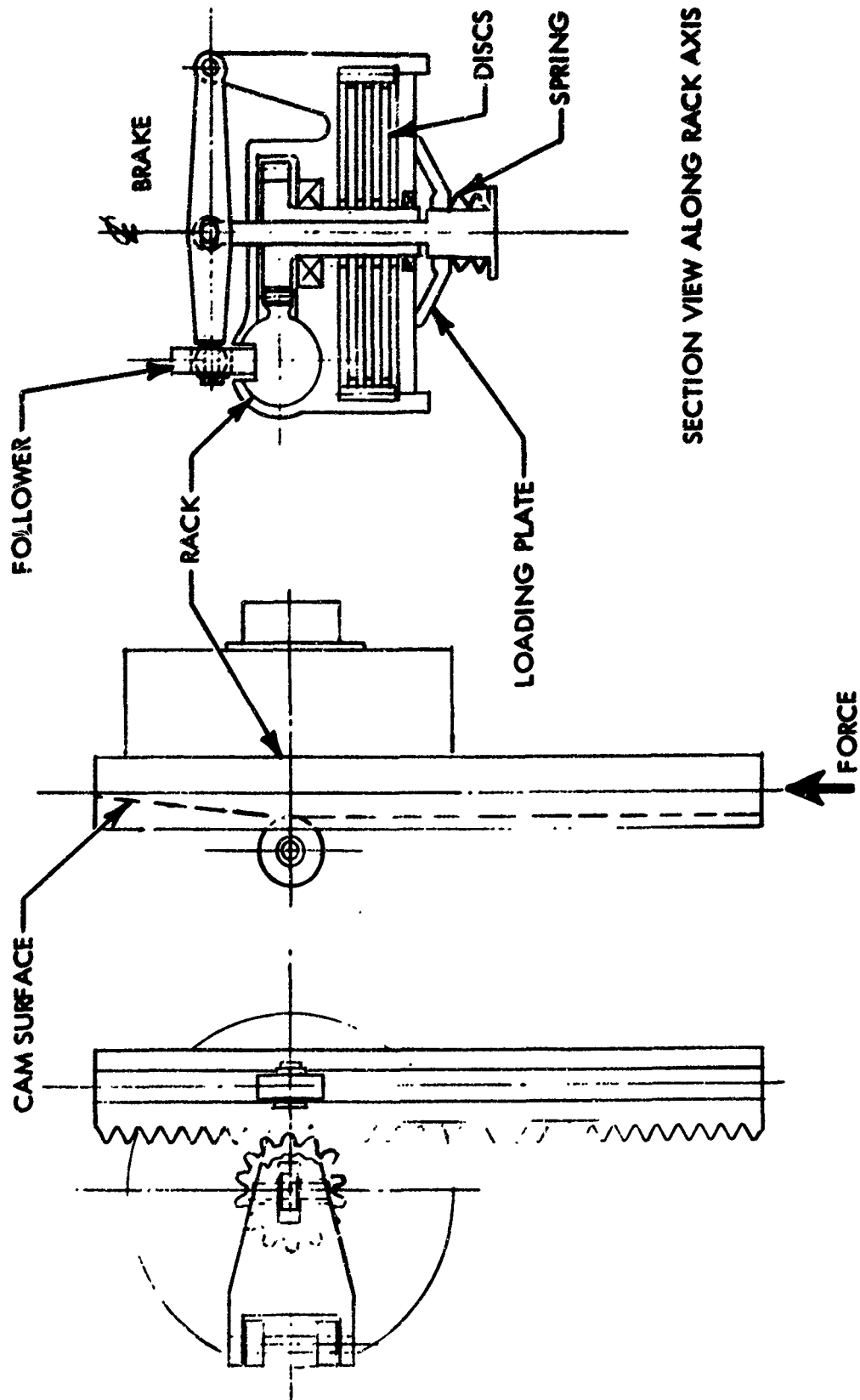


Figure 32. Rotary Brake - Rack and Pinion Actuated

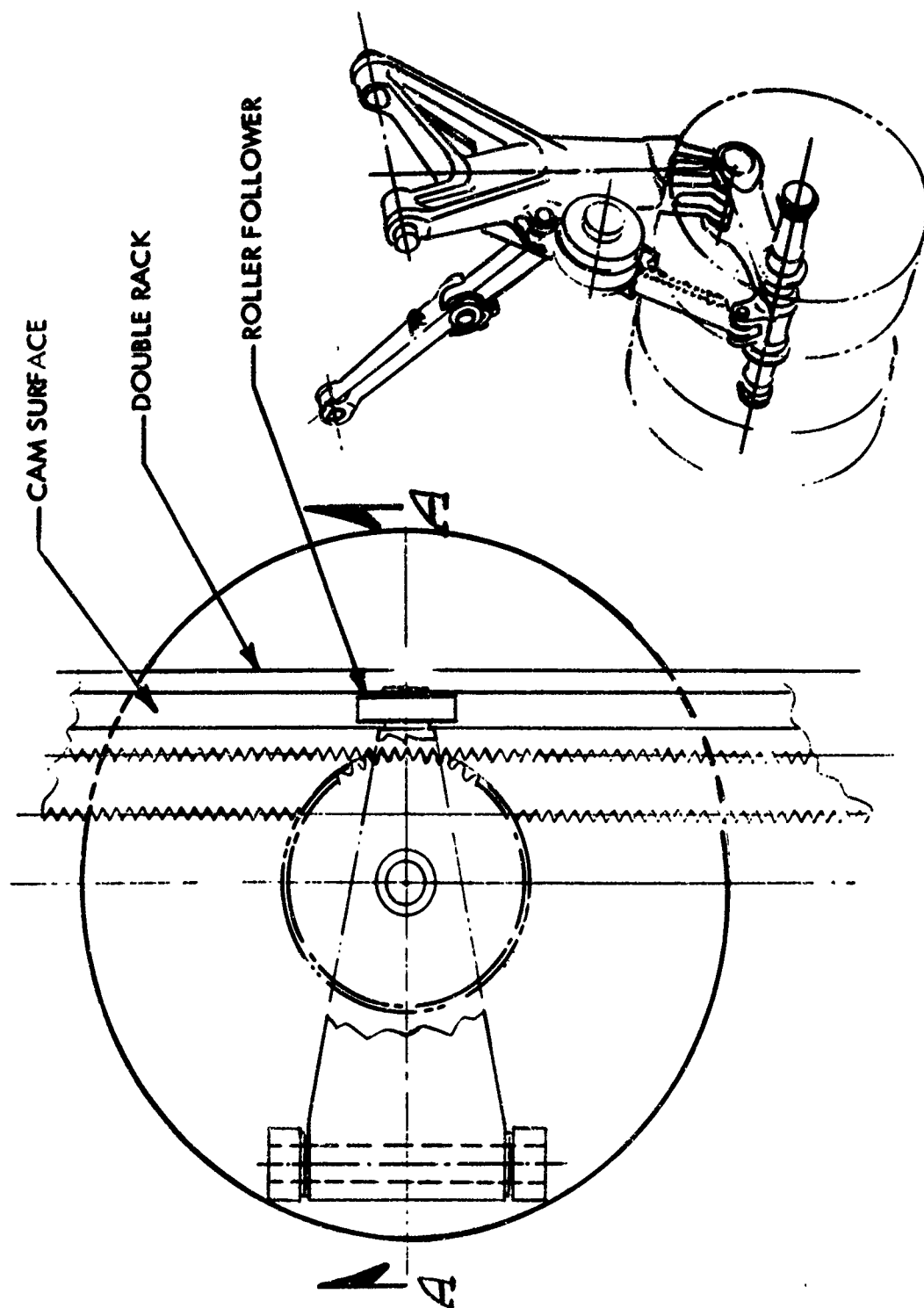


Figure 33. Centrifugal Control Disc Absorber (Sheet 1 of 2)

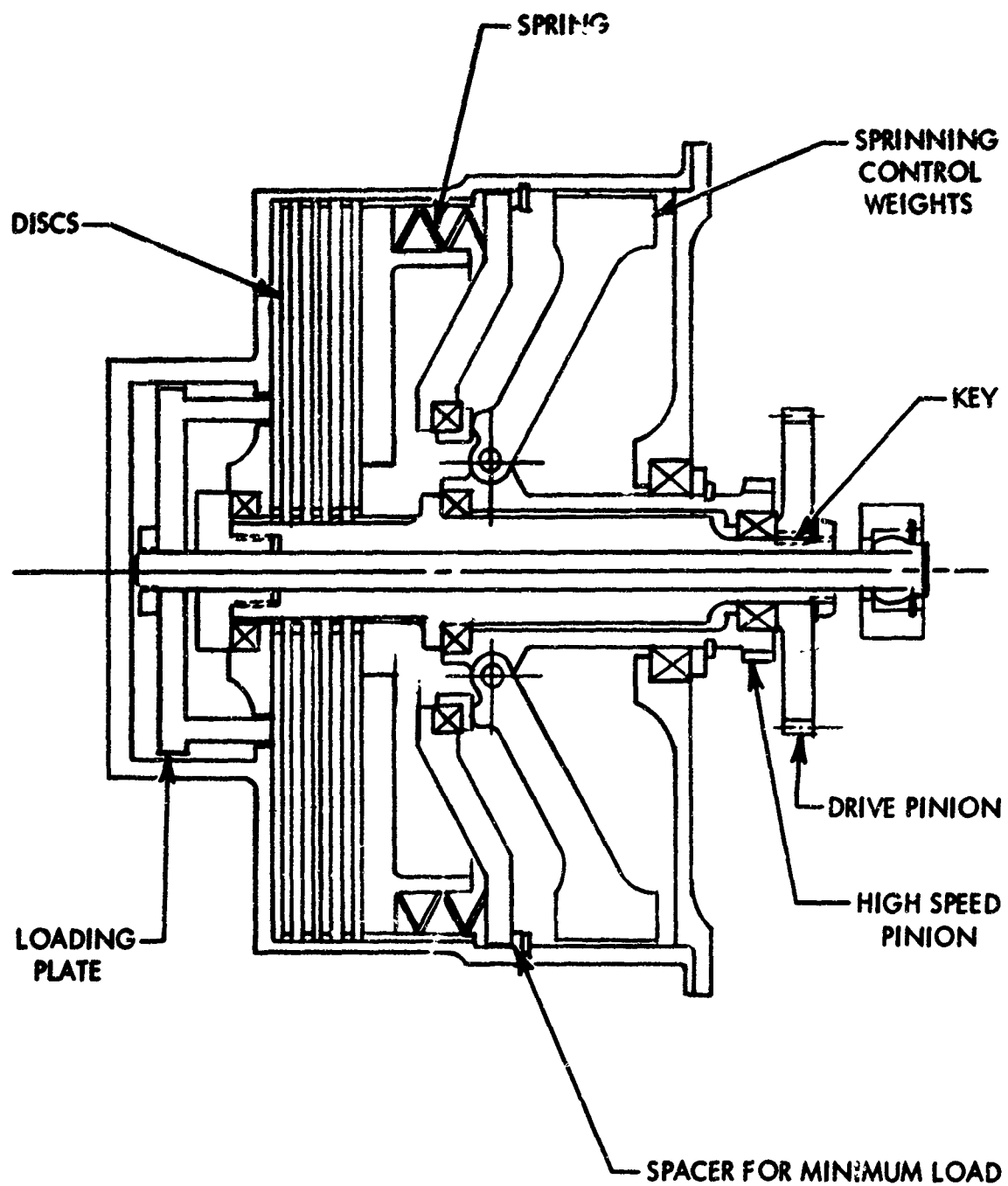


Figure 33. Centrifugal Control Disc Absorber (Sheet 2 of 2)
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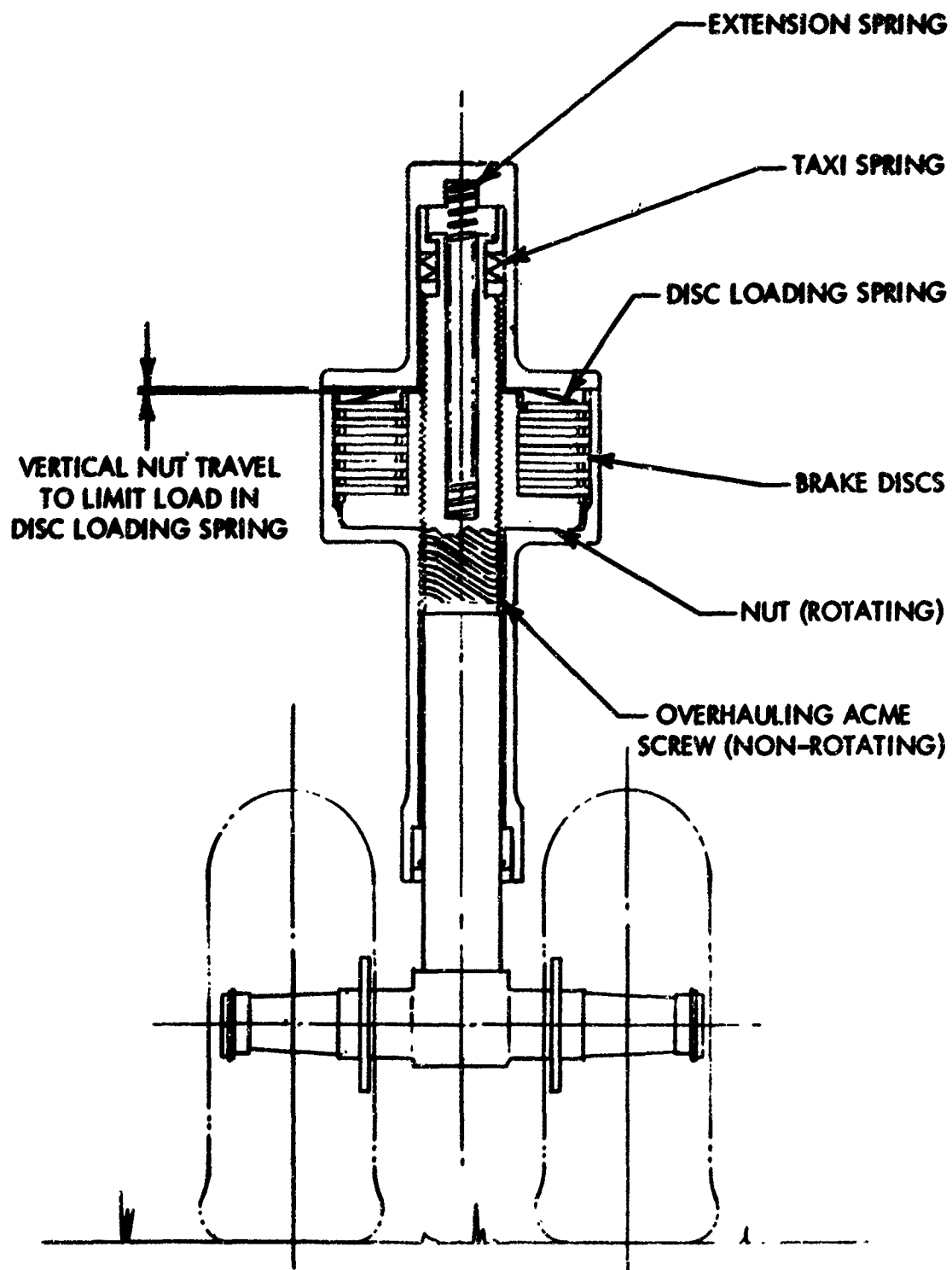


Figure 34. Rotary Absorber - Acme Screw

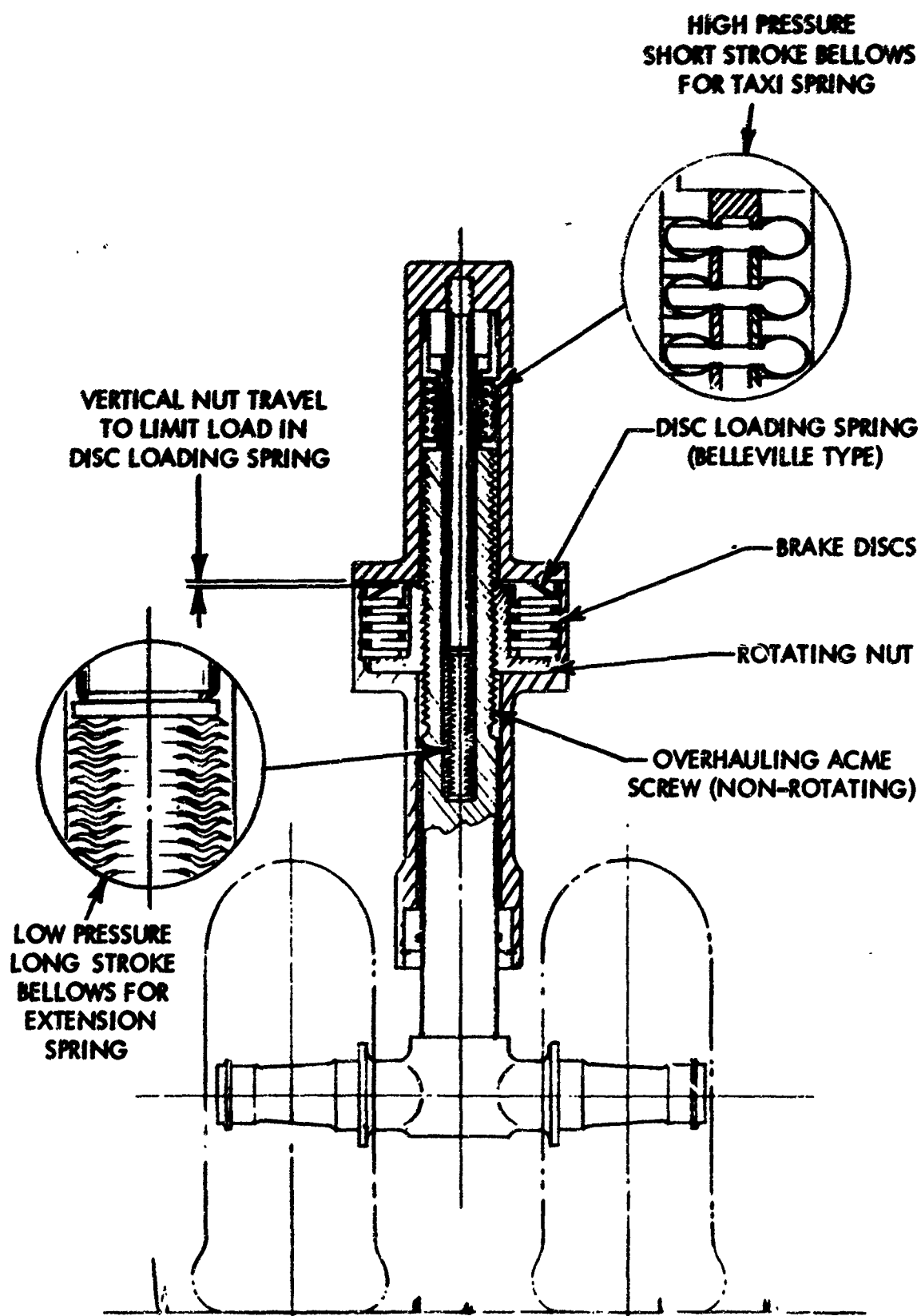


Figure 35. Rotary Absorber - with Metal Bellows

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APPENDIX B

Calculations Associated with a Linear Friction Shock Absorber

1.) Total Energy = $18.7 \times 14,600 = 273,000$ in-lb.

2.) Energy Absorbed by:

a.) Tire = 38,000 in-lb

b.) Bearings = 18,000 in-lb
($\mu = .13$)

c.) Balls = 35,000 in-lb
($\mu = .05$)

d.) Bellows = 44,000 in-lb
TOTAL: $\frac{135,000}{135,000}$ in-lb

Remaining K.E. = 138,000 in-lb

3.) Assume friction energy is dissipated in 8.25 inches(@ contact with bellows) & unit is 85% efficient strut load = $\frac{138,000}{.85 (8.25)} \approx 20,000$ lb.

4.) Find radial load on friction material to produce a 20,000 lb load based on $\mu = .20$

$$\frac{20,000}{.20} = 100,000 \text{ lb.}$$

5.) Unit loading = $\frac{100,000}{\pi \times 4.75 \times 3.50} \approx 2,000$ PSI

6.) Lining Power = $\frac{138,000 \text{ in-lb}}{12 \times 550 \times .20 \text{ SEC} \times 50 \text{ in.}^2}$
 $= 2.09 \frac{\text{HP}}{\text{IN}^2}$

7.) Ball Loading = $\frac{100,000 \text{ lb}}{54 \text{ balls}} = 1850 \text{ lb/ball}$

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APPENDIX C

Calculations Associated with an Overhauling Screw and Disc Brake Type Friction Shock Absorber

1.) Symbols and Data:

a.) Acme Thread - major diameter = 2.50 in.
pitch = 0.39 in.
lead = 1.95 in.
included angle = 29°

b.) $\mu = (.15)$ Coefficient of Friction (aluminum bronze to steel)

c.) F_a = Axial Load

d.) α = Helix Angle

e.) Θ = True Pressure Angle

f.) DM = Mean Diameter Thread

2.) Work of Friction =
$$\frac{\mu F_a P}{\sin \alpha (\cos \Theta + \mu \sin \alpha)}$$

3.) $\tan \alpha = \frac{P}{\pi DM} = \frac{1.95}{\pi (2.33)} = 0.266$

4.) $\sin \alpha = 0.258 \quad \tan \phi = \tan^{-1} 14.5^{\circ} = .258$

5.) $\cos \Theta = \frac{1}{(1 + \tan^2 14.5^{\circ} + \tan^2 \alpha)^{\frac{1}{2}}} = \frac{1}{(1.137)^{\frac{1}{2}}} = \frac{1}{1.065} = .938$

6.) $W_f = \frac{(.15) F_a (1.95)}{.258 (.938 + .15 (.258))} = \frac{(.15) F_a P}{.2519} = .60 F_a P$

7.) If the efficiency $e > 50\%$ the screw will overhaul.

$$e = \frac{26,100 (1.95)}{26,100 (1.95) + .65 (26,100) (1.95) + F_c \mu c \pi D_c}$$

Where D_c = Disc Dia. = 5.0 in.

F_c = Brake Disc force (lbs)

μ_c = Disc coefficient = 0.20

$$0.50 = \frac{50,895}{50,895 + 33,081 + F_c D_c \mu_c \pi}$$

$$F_c = \underline{5670 \text{ lbs max}}$$

8.) For Static Reactions

$$0.50 = \frac{14,600 \times 1.95}{14,600 \times 1.95 + .65 (14,600) (1.95) + 5 (.2) \pi F_c}$$
$$F_c = 3,172 \text{ lbs min}$$

9.) $F_c = \text{No. Weights} \times \text{Weight Load} \times \text{No. Friction Surfaces}$

$$= 4 \times P \times 8$$

$$P_{\max} = \frac{5670}{32} = 177 \text{ lb.}$$

$$P_{\min} = \frac{3172}{32} = 100 \text{ lb.}$$

Reactions required at governor lever and for determining centroid and weight of governor units -

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13. ABSTRACT		
<p>The purpose of this study is to evolve and evaluate a number of preliminary aircraft landing gear designs or concepts utilizing shock absorbers based on friction energy absorption methods adaptable to high temperature use (1000°F and up). Initial studies developed as many concepts as possible, without attempting to size components, but merely to portray an idea. Four (4) concepts were selected from the initial study and preliminary designs of the selected concepts were prepared based on landing and take-off characteristics of the USAF F-100 aircraft. Scale model drawings were then developed based on the full scale landing gear designs to experimentally evaluate the proposed designs. Data indicates that friction methods have extremely high energy absorption capacities, that the coefficient of friction is relatively insensitive to temperature variations and that a centrifugally controlled friction device has sufficient advantages to warrant further research effort.</p>		

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14. KEY WORDS	LINK A		LINK B		LINK C	
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Landing Gear - Design						
Landing Gear - Materials						
Landing Gear - Unconventional						
Landing Gear - Approach						

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